RFP-3291
UC-60

ENERTECH 2kW HIGH RELIABILITY WIND SYSTEM

Phase II - Fabrication and
Testing

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As a Part of the

UNITED STATES DEPARTMENT OF ENERGY
WIND ENERGY TECHNOLOGY DIVISION
FEDERAL WIND ENERGY PROGRAM

Contract No. DE-AC04-76DP03533

ABSTRACT

A high-reliability wind machine rated for 2kW in a 9 m/s wind has been developed by Enertech Corporation under Contract PF 64410 F with Rockwell International. Phase II of this contract centered on the fabrication and testing of prototypes of the wind machine. This report summarizes the Phase II activities performed by Enertech. The test results verified that the wind machine met the power output specification of the contract and that the variable-pitch rotor effectively controlled the rotor speed for wind speeds up to 50 mph. Three prototypes of the wind machine were shipped to the Rocky Flats test center in September through November of 1979.

Because of the high wind speeds needed to start the machine during tests at Rocky Flats, Enertech was awarded an additional contract in October of 1980 to reduce the start-up wind speed. At the completion of this contract, the start-up wind speed at the Enertech facility had been reduced to 4.5 m/s. A summary of the results of this program is included in this report.

PHASE II REPORT

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NOMENCLATURE LIST

AKWH - Annual Kilowatt-Hours Produced

amp - ampere

AOM - Annual Operation and Maintenance Cost

Ass'y - Assembly

C - Centigrade

cm - centimeter

COE - Cost of Energy

DC - Direct Current

E - Estimated failure rate based on best available generic data

E1 - Mean Failure Rate from Table 18.3, Generic Failure Rate Distributions, Mechanical Design and Systems Handbook, Harold Rothbart, ed., McGraw-Hill, 1964

E2 - Failure rate for 2-stage helical gear train assemblies from NACA Report #TR-824, available from Superintend of Documents

E3 - Failure rate value from NAVSHIPS 93820, Reliability Prediction Handbook

EHS - Extra High Strength

F - Fahrenheit

FDR - Final Design Review

FRC - Annual Fixed Charge Rate

hr - hour

Hz - Hertz

IC - Initial Installed System Cost

in - inch

kg - kilogram

kW - kilowatt

lb - pound

mm - millimeter

mph - miles per hour

m/s - meters per second

MTBF - Mean Time Between Failures

PDR - Preliminary Design Review

R - Reliability

REV - Revolution

rpm - revolutions per minute

t - time

VDC - Volts Direct Current

vs - versus

o - Degrees

" - Inches

λ - Failure rate

λ_c - Failure Rate of Component in the wind machine application

1. Introduction and Project Activities Summary

1.1 Introduction

Enertech Corporation is currently completing a development program to design, fabricate, and test a 2kW High-Reliability Wind Machine. The machine is to be capable of producing 2kW of electrical power in a 9 meter/second (20 mile/hour) wind and should require not more than one person-day of service per year.

Enertech is performing this development work under contract to Rockwell International as part of DOE's program to advance the technology and to accelerate the commercialization of reliable and economically viable wind energy systems. The applications envisioned for the machine are remote from other sources of electrical power and are widely dispersed in terms of geographical location and environment. Examples of applications include powering remote communications stations, supplying galvanic protection to buried pipelines, powering remote instrument systems, and powering off-shore or remote navigational aids.

Because of the very remote nature of many of these applications, the machine should be able to operate through a variety of extreme weather conditions and should be as easy to transport and install as possible.

The work described in this Phase II report is sponsored by the United States Department of Energy, Wind Systems Branch, and is administered by Rockwell International, Rocky Flats Plant.

Figure 1 shows the basic structure of the contractor organization. Enertech is the prime contractor to Rockwell and has one major subcontractor and one major consultant. Maremont Corporation of Saco, Maine, developed the electrical subsystem for the machine on subcontract. Their work included the development of the alternator, voltage regulator, and rectifier assemblies. Kaman Aerospace acted as consultants in certain areas of engineering. In particular Kaman has helped in doing reliability analyses and in doing rotor dynamics and aeroelastic analyses.

CONTRACTOR ORGANIZATION

DEVELOPMENT OF A 2KW
HIGH RELIABILITY
WIND MACHINE

CONTRACT NO. PF64410F

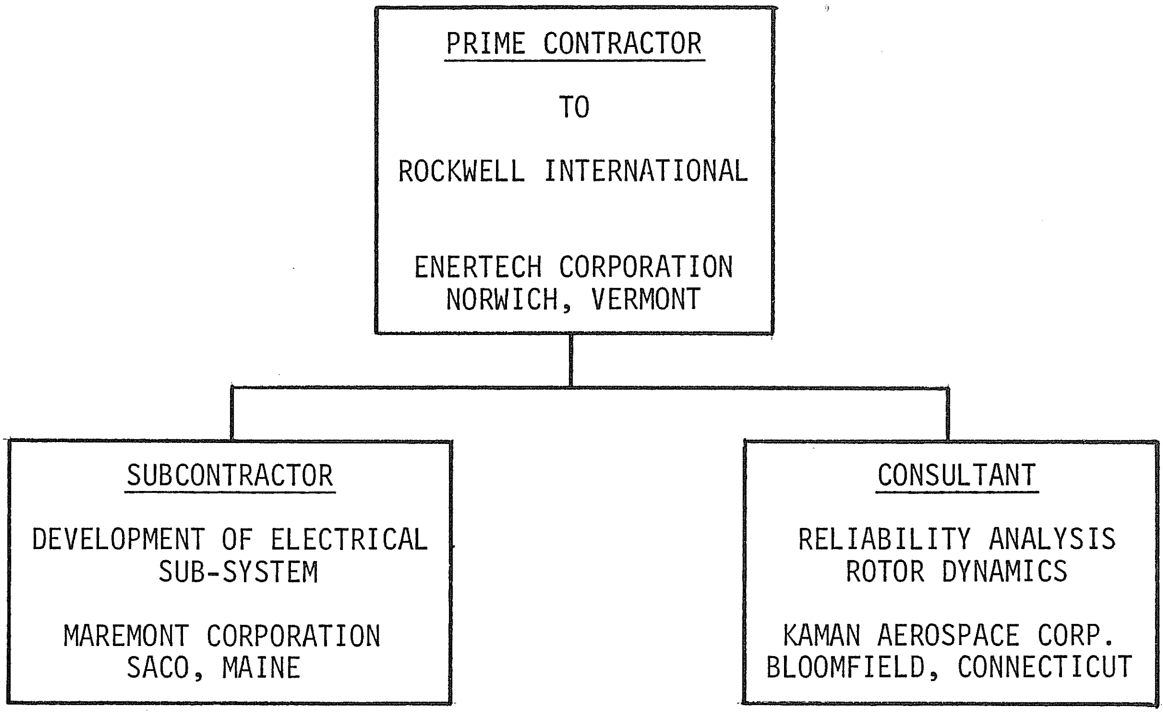

FIGURE 1: CONTRACTOR ORGANIZATION

The contract was awarded in January of 1978 and was scheduled to end in June of 1981. The work was divided into two portions. Phase I activities centered on the design and development of a 2kW prototype wind machine. Phase II activities centered on the fabrication and testing of prototype units.

The Phase I report was published in January of 1980. This Phase II report, the last item to be completed in the contract, summarizes the Phase II fabrication and testing activities.

1.2 Phase I Activities Summarized

During Phase I, a two-kilowatt wind machine was developed and the concept of the design was tested. Figure 2 shows the Phase I contract schedule and some of the major milestones in the program.

Preliminary Design Review, PDR, was held April, 1978. At that time, two design concepts were proposed.* Reliability studies had been conducted for both designs. Following PDR, preparation for Critical Design Review, CDR, included detailed stress analysis of both concepts. To meet the design specifications, a variable-pitch hub, utilizing a torsion bar spring system, was chosen. Detailed rotor design, including drawings and stress analysis, was completed for CDR, held September, 1978. The hub design submitted at CDR was fabricated and tested. Improvements were made to the design as needed. At Final Design Review, held December, 1978, drawings for the entire wind generator were submitted. Further truck testing was performed through January and February of 1979. Again, improvements to the rotor were specified. At a final wrap-up meeting held March 1, 1979, final production drawings were submitted to Rocky Flats.
The Phase I report summarizing Phase I activities was printed in January of 1980 and is available through the National Technical Information Service.

1.3 Phase II Activities Summarized

The generalized schedule of Phase II activities is shown in Figure 3.

* For a description of the two proposed designs and subsequent trade-off analyses, refer to Development of a 2 Kilowatt High Reliability Wind Machine, Phase I - Design and Analysis, January 1980, W. Drake, et al., Enertech Corporation, Volume I - Executive Summary (RFP-3025-1), Volume II - Technical Report (RFP-3025-2). (Both are available from NTIS)

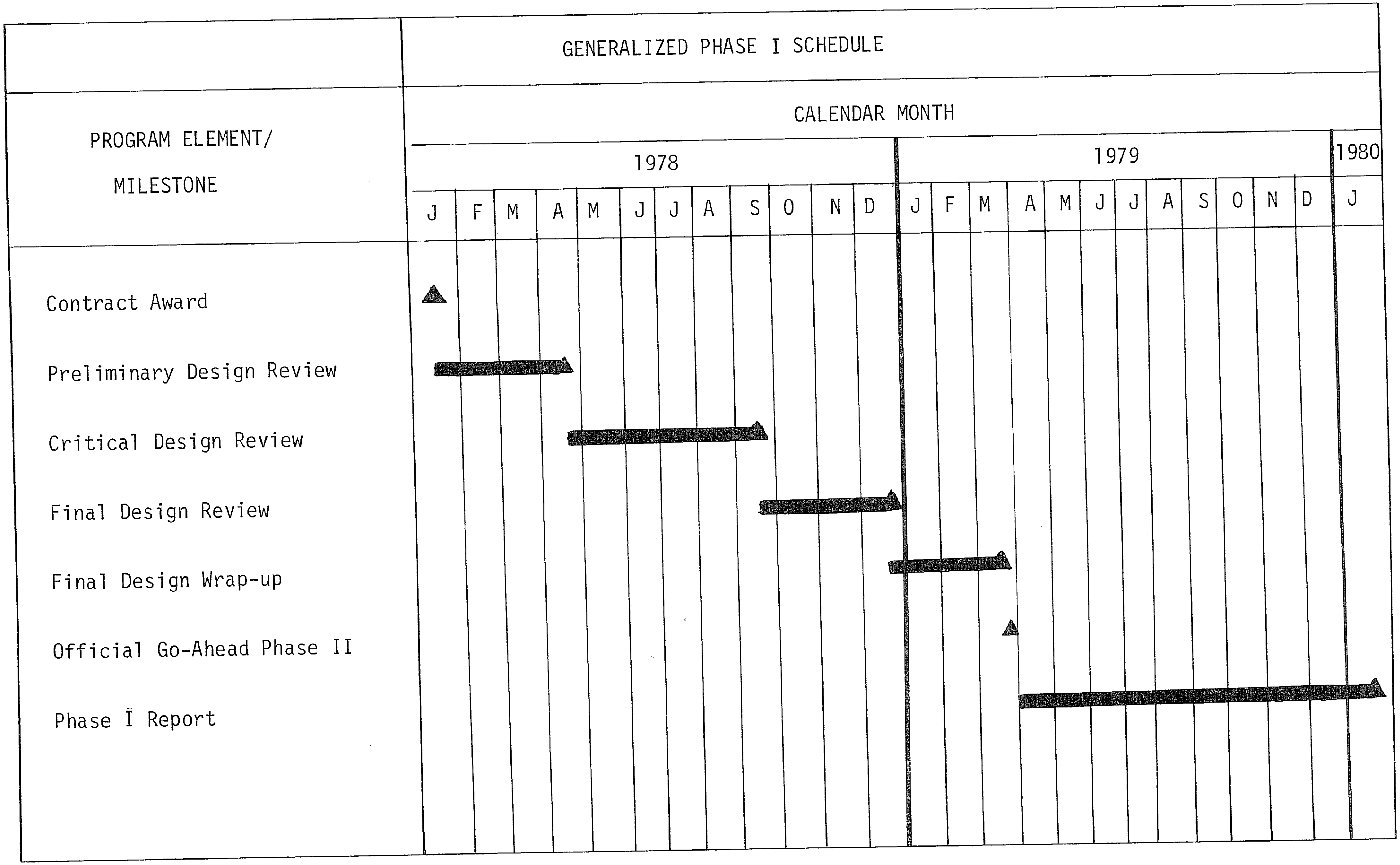

FIGURE 2: GENERALIZED PHASE I SCHEDULE

GENERALIZED PHASE II SCHEDULE

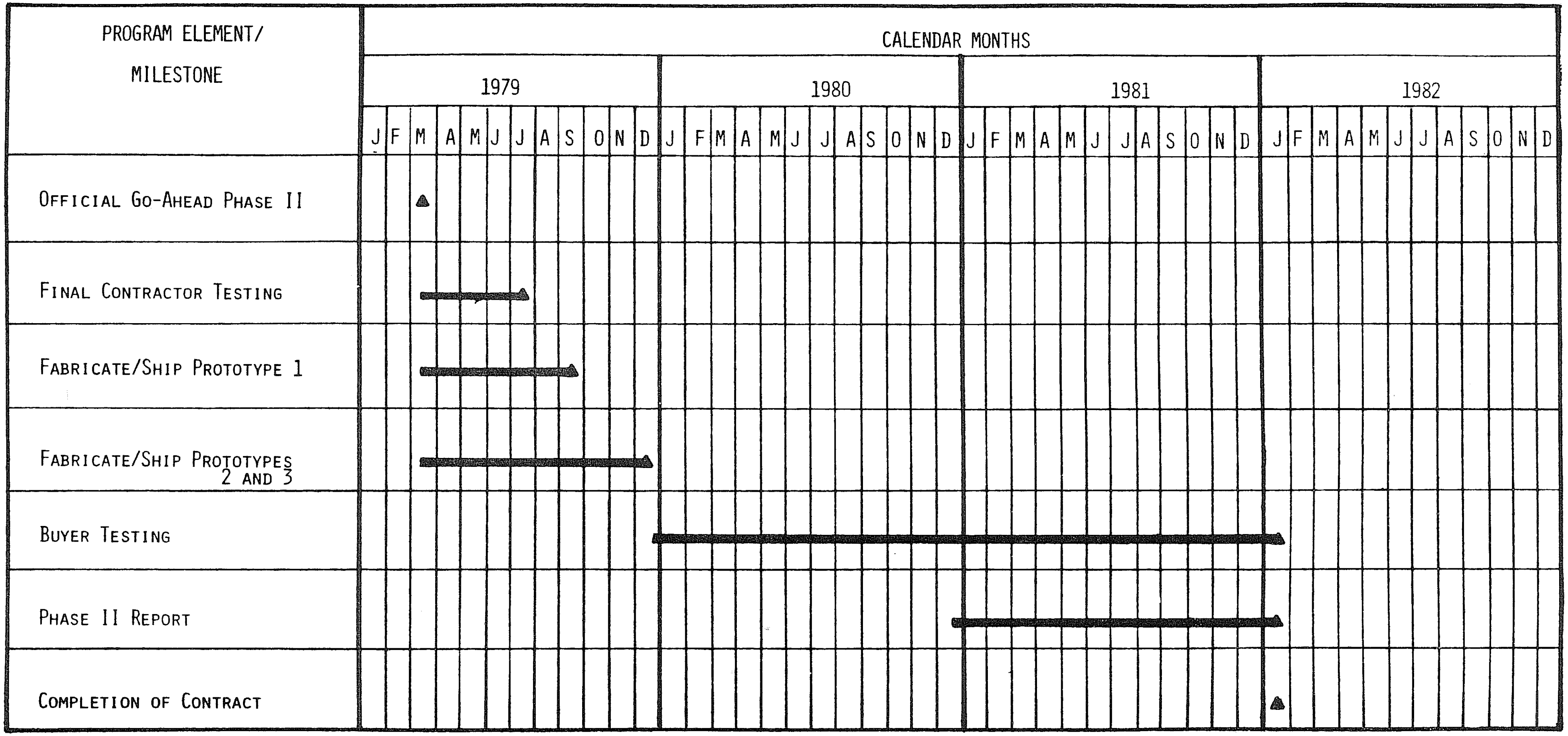

FIGURE 3: GENERALIZED PHASE II SCHEDULE

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In March of 1979, Enertech Corporation received the official go-ahead for Phase II activities. Final contractor testing under this contract took place between March and July of 1979. Following the final atmospheric testing at Mount Washington in New Hampshire, the wind machine was disassembled for shipment, crated, and sent to Rocky Flats in September of 1979 for Rocky Flats testing. Prototypes 2 and 3 were shipped to Rocky Flats in November. Testing at the Wind Systems Test Center began in the fall of 1979 and continued through the close of this contract. Figure 4 shows the schedule of Phase II activities.

In October of 1980 the Enertech Corporation accepted a fixed-price contract to improve the start-up capabilities of the wind machine. Although the work on the fixed-price contract is not included in the Phase II schedule, shown in Figures 3 and 4, the work will be discussed in the testing portion of this report.

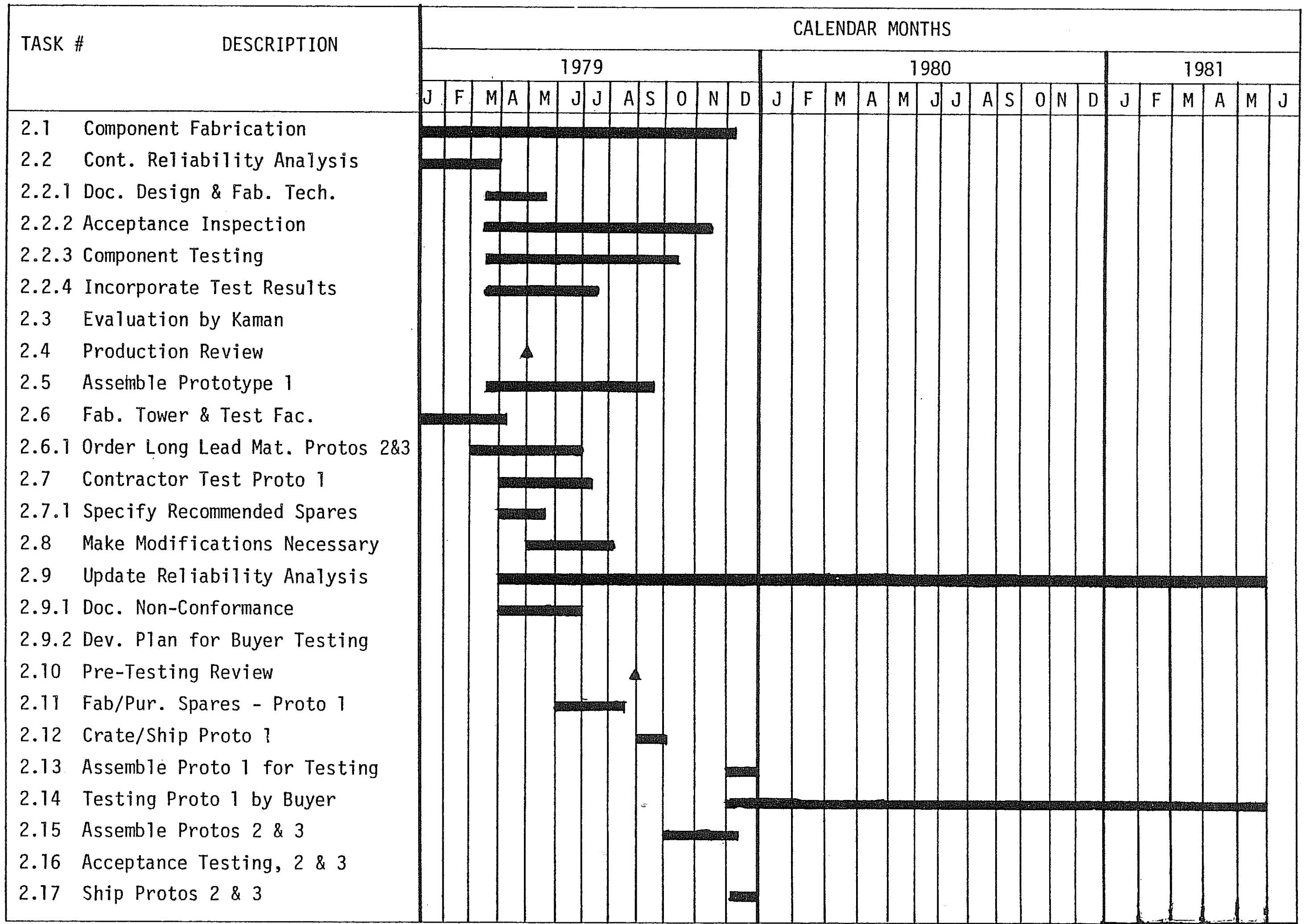


FIGURE 4: SCHEDULE BY TASK OF PHASE 2 ACTIVITIES

2. Overview of the Design

2.1 Introduction

Figure 5 summarizes the design specifications for the 2kW wind machine developed under this contract. The unit is designed to produce 2kW of electrical power in a 9 m/s wind, to operate under a variety of environmental conditions, to have a design life of 25 years, to require only one maintenance day per year, and to have a mean time between failures of at least ten years.

Figure 6 shows the configuration of the wind machine on a tower. The machine is a downwind, horizontal-axis wind machine with two wooden blades. The rotor diameter is 5 meters. Rotor speed control is accomplished by pitching the blades into a stalled condition. The pitching of the blades is caused by the centrifugal force on weights attached near the blade roots. The tower recommended for the machine is a 40-foot guyed communications tower.

Figure 7 summarizes a number of the technical characteristics of the machine. The power output of the machine is 3.0 kW at 9 m/s and the maximum power output is 3.7 kW in a 13.5 m/s wind. The output from this machine is nominally 24 volts DC. The machine operates at all wind speeds above the start-up wind speed with the rotor speed being governed at wind speeds above approximately 13.5 m/s. The rotor operates through a speed range from zero to 350 rpm with power being produced at speeds above 150 rpm.

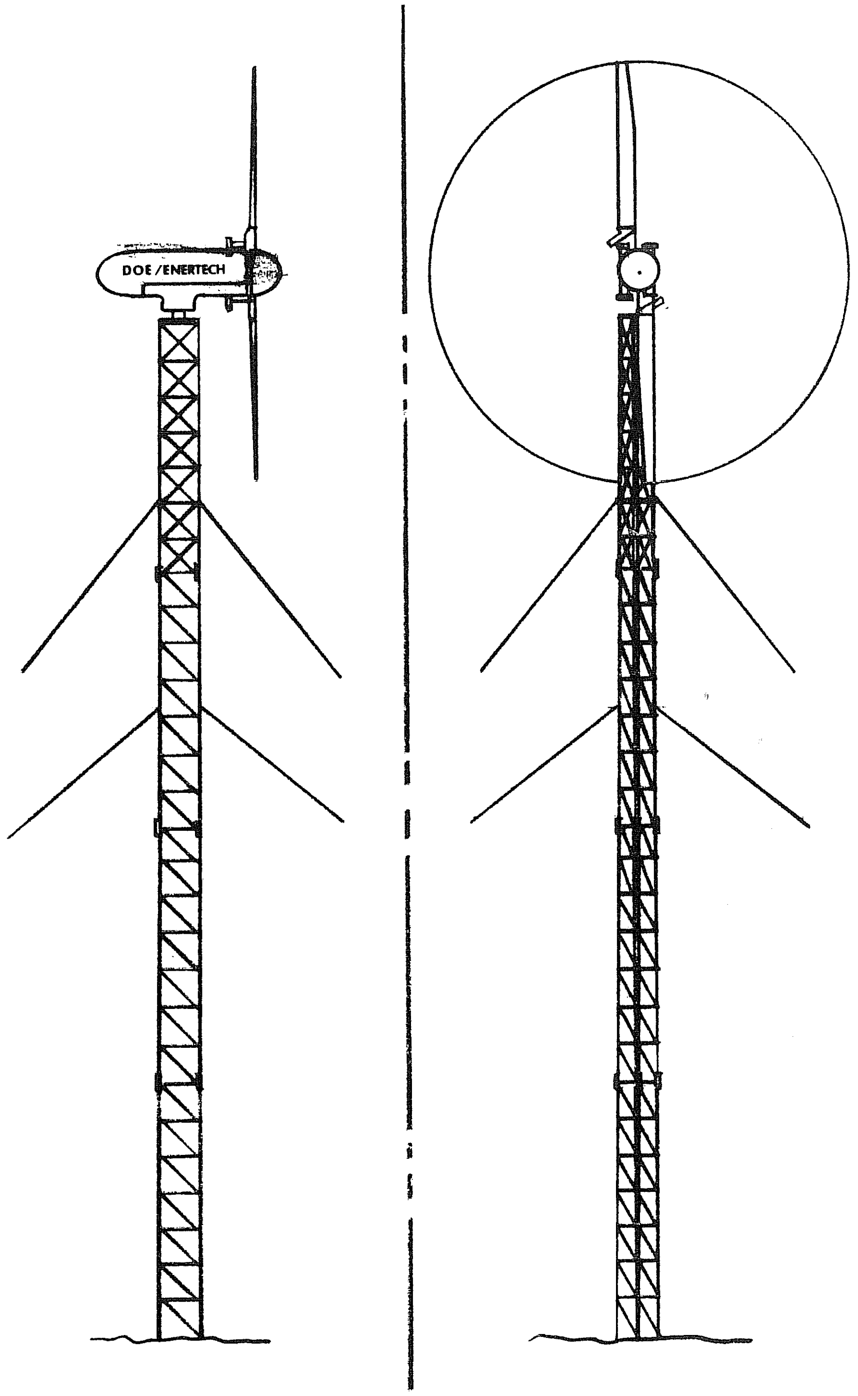
2.2 Rotor Design

Figure 8 shows a drawing of the rotor hub. This drawing shows the hub in a view looking down the axis of rotation of the rotor. The blades are attached at the top and bottom of the drawing and extend out of the drawing. The right side of the drawing is shown in a normal view; the left side shows a cut-away view.

FIGURE 5

DESIGN SPECIFICATIONS

POWER OUTPUT	2kW (MINIMUM) AT HUB HEIGHT @ 9 m/s
TEMPERATURE	-70° to 60°C (-94°F to 140°F)
RAIN	TORRENTIAL DOWNPOUR ACCOMPANIED BY WINDS
SNOW, SLEET, ICING	ICE BUILD-UP TO 60mm (2.5 in.) THICK
HAIL	IMPACT BY 40mm (1.5 in.) HAIL
SALT WATER SPRAY	HEAVY OCEAN SPRAY
CORROSIVE ATMOSPHERE	HEAVY INDUSTRIAL ATMOSPHERE WITH SALT FOG AND SPRAY
DUST	FINE SAND AND DUST ACCOMPANIED BY PERSISTANT WINDS TO 45 m/s (100 mph)
WIND	54 m/s (120 mph) STEADY WIND, GUSTS TO 75 m/s (165 mph)
LIGHTNING	REPEATED STRIKES DURING THUNDERSTORMS
NOISE	MAXIMUM 50 dbA MEASURED AT THE BASE OF A 40' TOWER IN A 9 m/s (20 mph) WIND
SYSTEM LIFE	25 YEARS MINIMUM
MAINTENANCE	1 PERSON-DAY PER YEAR MAXIMUM
RELIABILITY	MEAN TIME BETWEEN FAILURES NOT LESS THAN TEN YEARS



ENERTECH 2KW HIGH RELIABILITY WIND MACHINE ON 40 FOOT TOWER

FIGURE 6

FIGURE 7

SUMMARY OF TECHNICAL CHARACTERISTICS OF 2kW HIGH-RELIABILITY WIND MACHINE

SYSTEM:	
BUS BAR POWER OUTPUT AT RATED WIND SPEED OF 9 m/s (20.1 mph)	3.0 kW
ALTERNATOR OUTPUT AT RATED WIND SPEED OF 9 m/s (20.1 mph)	3.0 kW
COEFFICIENT OF PERFORMANCE, Cp, AT RATED WIND SPEED	.34
MAXIMUM POWER OUTPUT - WIND SPEEDS OVER 13.5 m/s (30 mph)	3.7 kW
OUTPUT VOLTAGE, NOMINAL	24 VDC
MINIMUM WIND SPEED TO START ROTOR	4.5 m/s (10 mph)
CUT-IN WIND SPEED (150 rpm ROTOR)	3.5 m/s (7.8 mph)
INITIATE GOVERNING WIND SPEED (APPROXIMATE)	13.5 m/s (30 mph)
CUT-OUT WIND SPEED	NONE (UNIT OPERATES AT ALL WIND SPEEDS)
SURVIVAL WIND SPEED	75 m/s (165 mph)
SYSTEM LIFE	25 YEARS (MIN)
MAINTENANCE REQUIREMENT	1 DAY PER YEAR (MAX)
SYSTEM WEIGHT (LESS TOWER)	310 kg (685 lbs)
ROTOR:	
TYPE	HORIZONTAL AXIS, DOWNWIND
NUMBER OF BLADES	2
ROTOR DIAMETER	5 METERS (16.4 FT.)
BLADE MATERIAL	SOLID WOOD, 8 mil POLYURETHANE LEADING EDGE PROTECTION
BLADE AIRFOIL SECTION	FLAT-BOTTOM AIRFOIL
BLADE ROOT CHORD	25 cm (10 in) ROOT, 18 cm (7.0 in) TIP

FIGURE 7 CONTINUED

BLADE THICKNESS	25% AT ROOT, 10.7% AT TIP, LINEAR TAPER
BLADE TWIST	6°
ROTOR SOLIDITY	.048
<u>SYSTEM</u>:	
ROTOR DESIGN TIP SPEED RATIO	6:1
MAXIMUM ROTOR SPEED	350 rpm
ROTOR PRE-CONE ANGLE	0°
ROTOR AXIS TILT FROM HORIZONTAL	0°
BLADE WEIGHT	6.8 kg (15 lbs)
HUB WEIGHT	84 kg (186 lbs)
ROTOR WEIGHT - TOTAL	98 kg (216 lbs)
<u>CONTROLS</u>:	
HIGH WIND OVERSPEED	CENTRIFUGALLY PITCHING HUB STALLS BLADES
VOLTAGE REGULATION	ADJUSTABLE SOLID-STATE VOLTAGE REGULATOR
ROTOR BRAKE (FOR MAINTENANCE SHUTDOWN)	MECHANICAL DISK AND CALIPER (GROUND-OPERABLE BY CABLE)
YAW BRAKE (FOR MAINTENANCE SHUT-DOWN)	NONE
<u>GEARBOX</u>:	
TYPE	TWO-STAGE, HELICAL GEAR
RATIO (PROPELLER TO GENERATOR)	1:10.9
LUBRICATION	WET SUMP, SYNTHETIC GEAR OIL
WEIGHT	37 kg (82 lbs)
<u>GENERATOR</u>:	
TYPE	TWO-PART, THREE-PHASE, WOUND FIELD ALTERNATOR
OUTPUT AND SPEED AT 9 m/s RATED WIND SPEED	3.0 kW at 2850 rpm
MAXIMUM OUTPUT AND SPEED	3.7 kW at 3500 rpm

FIGURE 7 CONTINUED	
WEIGHT	30 kg (65 lbs)
SYSTEM:	
TOWER:	
TYPE	ROHN 45GSR
HEIGHT	40 FEET
WEIGHT	375 kg (825 lbs)

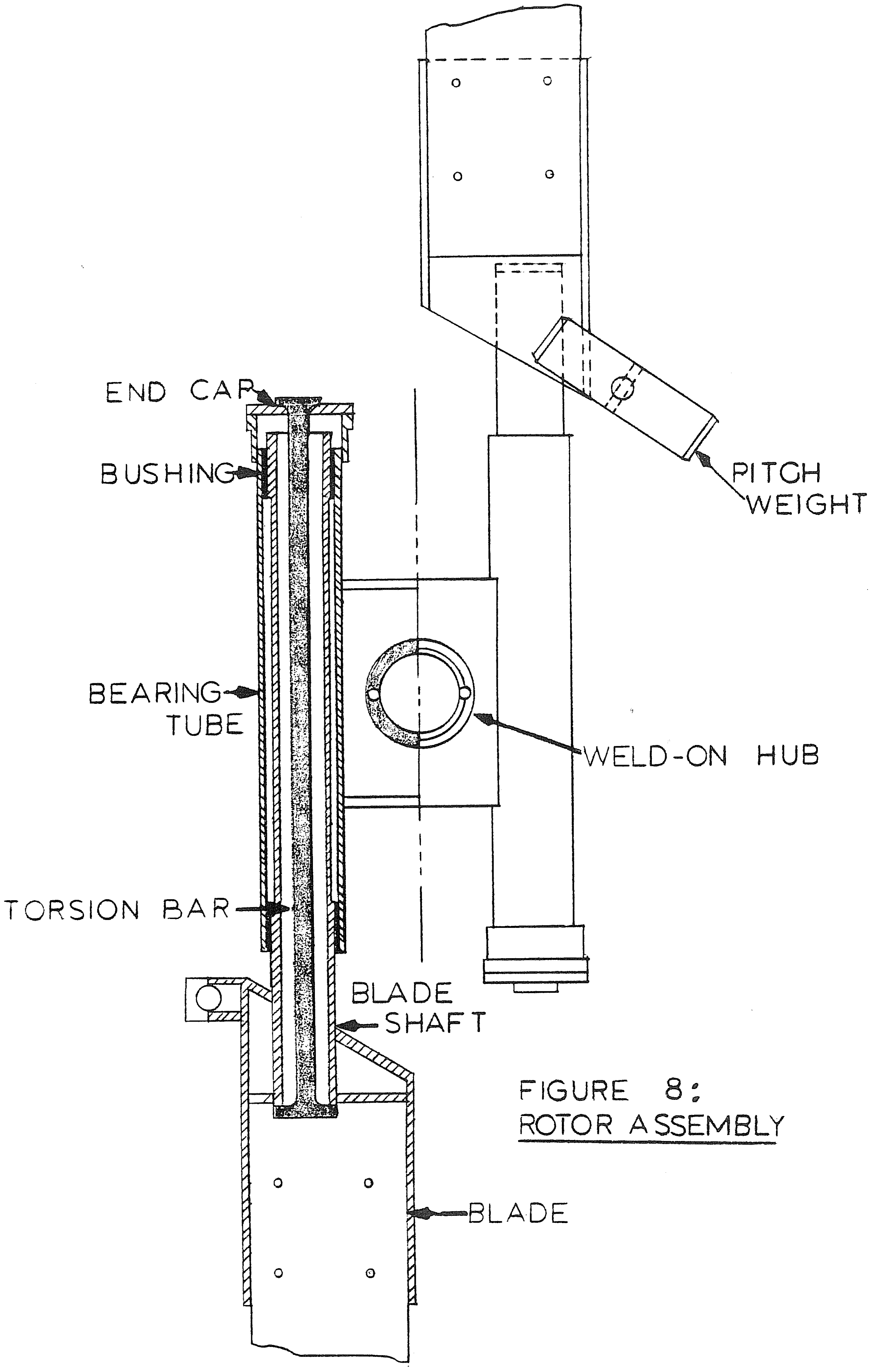


FIGURE 8: ROTOR ASSEMBLY

The solid wood blades are made from sitka spruce. The blade chord tapers from 10 inches at the root to 7 inches at the tip. The weight of one blade is approximately 15 pounds.

Each blade is attached to a steel blade shaft assembly which is supported inside a hub bearing tube by two teflon bushings. These bushings allow the blade shaft and blade to pitch. The blade shaft is retained in the hub by a stainless steel tension/torsion bar running down the center of the blade shaft. The tension/torsion bar is attached to the hub bearing tube at one end and to the blade shaft at the other end. This bar resists the centrifugal loads on the blade and blade shaft and acts as a torsional spring resisting the pitching of the blades. When the rotor spins, centrifugal force on the pitch weights produces the moments which pitch the blades. The blades pitch toward stall to control the rotor speed. The maximum pitch travel for this machine is approximately 19 degrees.

2.3 Drive Train

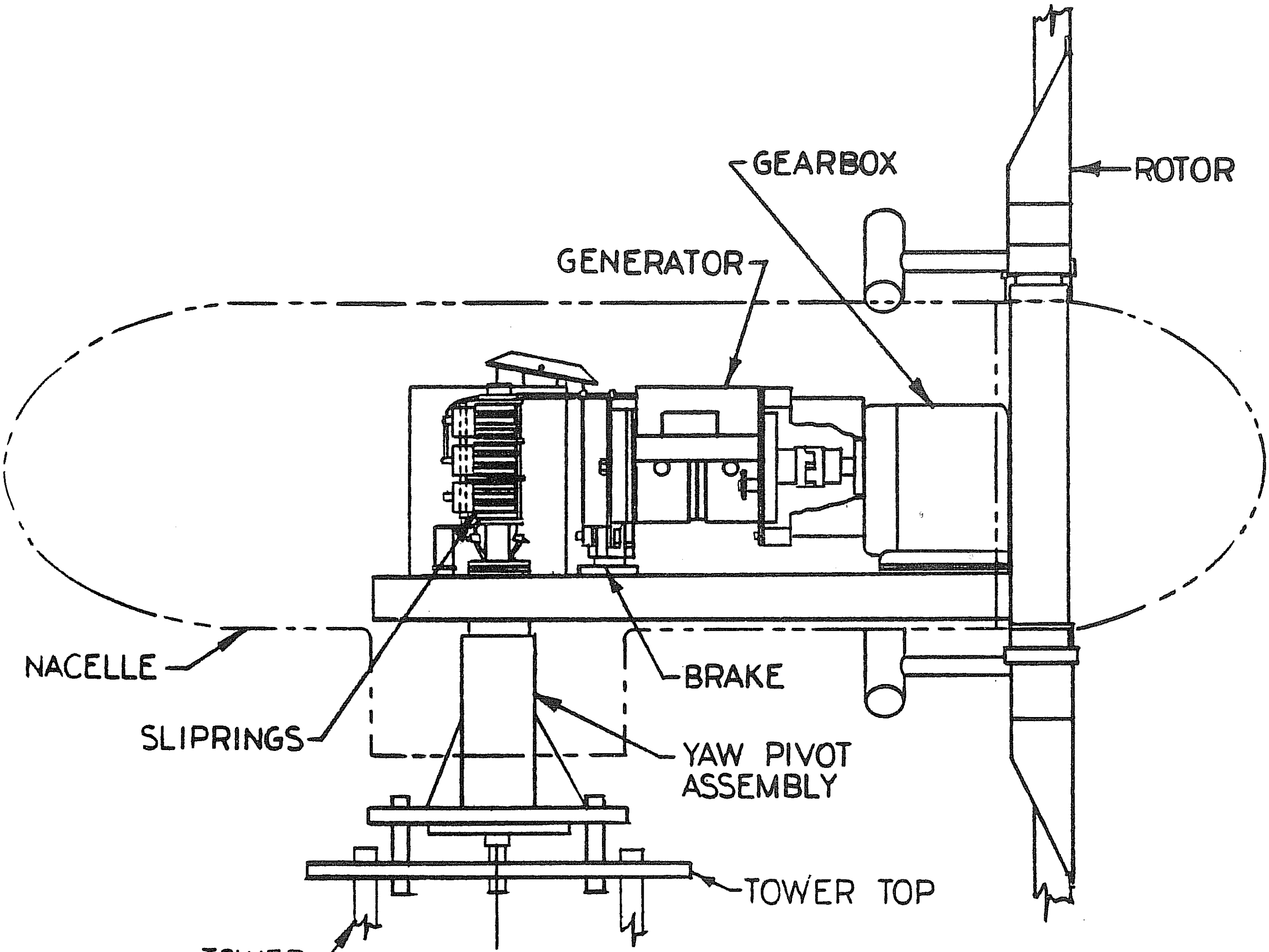
The drive train consists of a gearbox, an alternator, and a rotor brake. The main body of the machine is shown in Figure 9. The rotor is on the right-hand side of the drawing.

The gearbox is a double-reduction helical gearbox with a step-up ratio of one to 10.9 from the rotor to the alternator. The rotor mounts directly to the low-speed shaft of the gearbox by means of a tapered bushing and utilizes the gearbox structure and bearings for support.

The alternator is a three-phase brushless alternator developed for this contract by Maremont Corporation under subcontract to Enertech. The alternator is a special design to improve the reliability of the system. Essentially, the alternator assembly consists of two separate alternators sharing the same shaft and housing. The two sections of the alternator have separate rectifiers and voltage regulators mounted on the housing. The alternator is mounted on the gearbox by a cast aluminum adaptor and is coupled to the high-speed shaft of the gearbox by means of a steel/bronze coupling.

FIGURE 9

ENERTECH 2KW WINDMACHINE ASSEMBLY



The rotor brake is designed to be used during maintenance of the machine. The brake is a floating-caliper disk brake with the stainless steel brake disk mounted on the alternator shaft. The brake is actuated from the base of the tower by a cable and linkage system.

2.4 Supporting Frame

The drive train components are supported on a steel frame that is part of the main frame/yaw pivot assembly shown in Figure 10. The yaw pivot consists of a vertical shaft welded to the main frame and a housing which mounts to the tower top. The shaft rotates within the housing on two radial roller bearings and a needle thrust bearing. The machine is free in yaw.

2.5 Electrical System

Electrical connection between the rotating and stationary parts of the yaw pivot is provided by slip rings located on the yaw axis. The machine has three slip rings: two for the main DC power from the alternator and one to ground the machine for lightning protection. Double brushes on each slip ring increase the reliability of the wind machine. The slip rings are protected from moisture and debris by a clear plastic case.

Lightning protection is provided by surge protectors on the slip rings and in the control box. The frame is grounded through the ground slip ring and the tower is grounded to grounding rods at the base of the tower and at the guy wire anchors.

2.6 Nacelle

In addition to protective coatings on the machine components, the wind machine is protected from the environment by a fiberglass nacelle that encloses the machine.

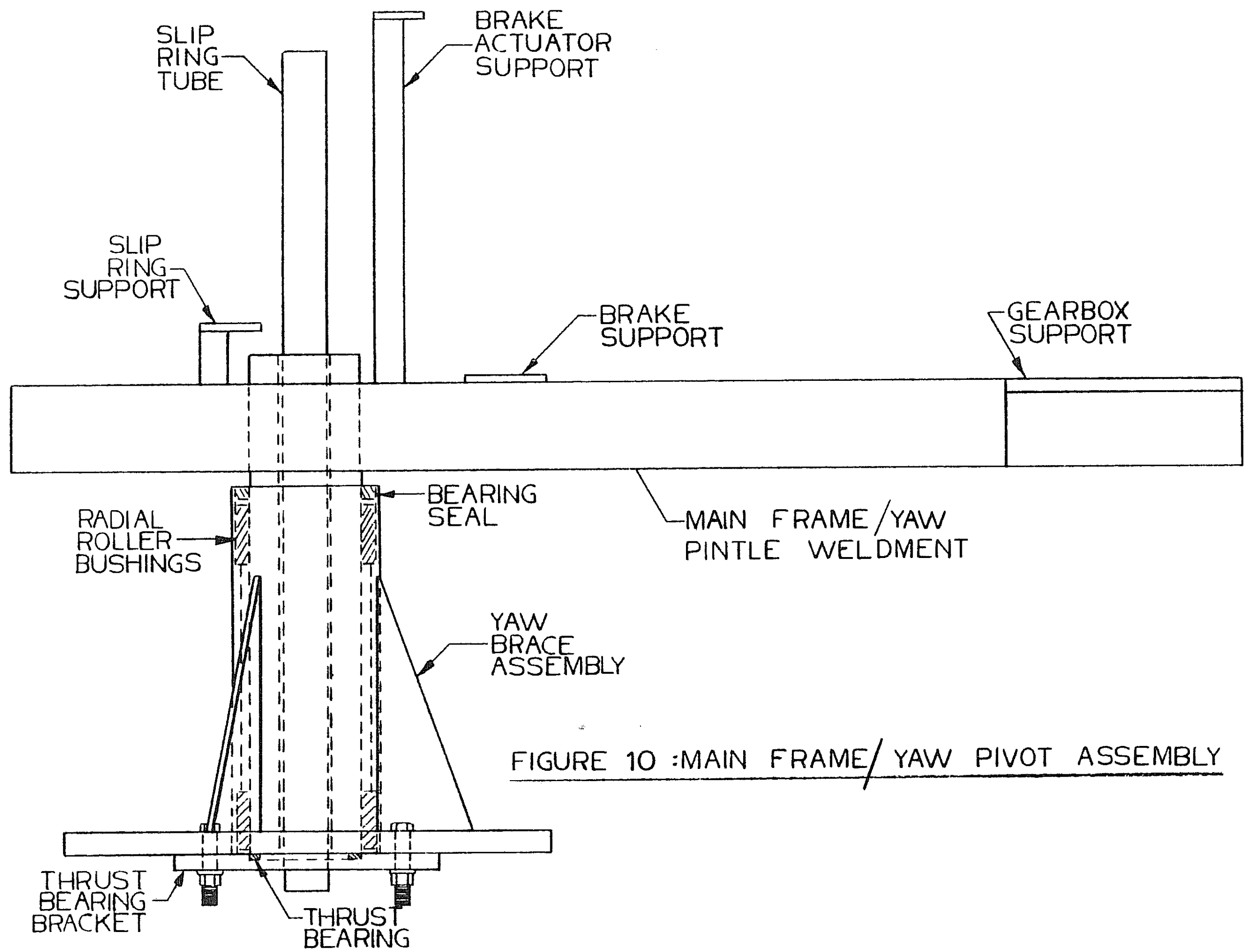


FIGURE 10 :MAIN FRAME/ YAW PIVOT ASSEMBLY

2.7 Tower

The tower recommended for this wind machine is a 40-foot Rohn 45GSR communications tower. The tower is designed to support the machine in a 75 m/s wind. The tower has three legs with cross-bracing between the legs. The top ten feet of the tower has double cross-bracing. The tower is guyed 24 feet and 31 feet above its base with three guy wires at each level. The entire tower is hot-dip galvanized for corrosion protection.

3. Design Changes During Phase II

3.1 Introduction

Several design changes of the baseline design (the design specified at Final Design Review) were required as a result of the Phase II fabrication and test programs. Major changes were specified for the blade pitching mechanism, the blades, the yaw pivot assembly, the fastener locking mechanism, the lightning protection system, and the tower. Minor changes were specified for several other components to correct problems encountered during testing, to reduce the fabrication cost, or to reduce assembly time.

3.2 The Rotor

During contractor truck tests in June of 1979, the rotor speed peaked at 380 rpm. Changes were necessary to limit the maximum rotor speed to less than 350 rpm. The initial blade root pitch (the pitch when the rotor is stopped) was decreased from 10^{o} to 9^{o} and the mass of the pitch weights was increased slightly.

Because the rotor speed had exceeded the design speed during the truck tests, the torsion bars were inspected for signs of fatigue damage. Dye penetrant tests revealed what were believed to be circumferential cracks in the welds. A fatigue analysis showed that the welds had only marginal fatigue strength. Because of this, the torsion bars were redesigned with 5/8" rather than 3/8" fillet welds to improve their fatigue strength. A new set of torsion bars with the 5/8" welds was fabricated for Prototype 1 to replace the ones with the cracks. The redesigned torsion bars were also used on Prototypes 2 and 3. It should be noted that subsequent tests at Rocky Flats revealed that the cracks were not actually fatigue cracks but were actually joints where the fabricator had left out welds which were specified in the drawings. Although there was not an actual failure of the welds during the test, the increased weld size is still thought to be desirable because the fatigue analysis showed the old design to be marginal.

The blades were redesigned in December of 1980 as part of the fixed-price contract. The blade material was changed from a white ash/spruce composite to a lighter weight sitka spruce blade. The airfoil shape was changed to a flat-bottomed airfoil. The chord at the blade root was increased from 9 to 10 inches. The initial blade root pitch was increased from 9 to 9.5 degrees.

During controlled velocity tests at Enertech, the redesigned blades were compared with the original design. For both designs, start-up began in 4.5 m/s winds. The power output at the 9 m/s rated condition was 3kW for the redesigned blades and 2.2kW for the original blades. For both blade sets, the maximum rotor speed was limited as desired.

The rotor with the redesigned blades was not re-analyzed for aeroelastic stability. The original design was thought to be sufficiently far from an unstable condition that small design changes could be made without causing problems. No evidence of aeroelastic instability was observed during controlled-velocity tests with the redesigned blades.

The calculated value of the fundamental flatwise bending frequency of the blades varies from 15.6 Hz. for a stopped rotor to 17.7 Hz. for a rotor turning at 350 rpm. Because of the variability of wood, some variation of the natural frequency from blade to blade is expected. The natural frequency of the redesigned blades is similar to the frequency of the original blades.

For easier fabrication and assembly, several other changes were made to hub parts. The groove angle of the welds in the hub weldment was decreased from 45 to 30 degrees. The size of the back-up plates for the blades was decreased slightly. A torsional locking screw, shown in Figure 11, was added to "lock" the initial blade pitch.

3.3 The Yaw Pivot Assembly

In April of 1979, the wind machine was mounted on a tower at Enertech to operate in a free-yaw condition. Yaw motion during the test was noticeably sluggish. For the baseline design, the minimum torque necessary to yaw the machine was 40 foot-pounds.

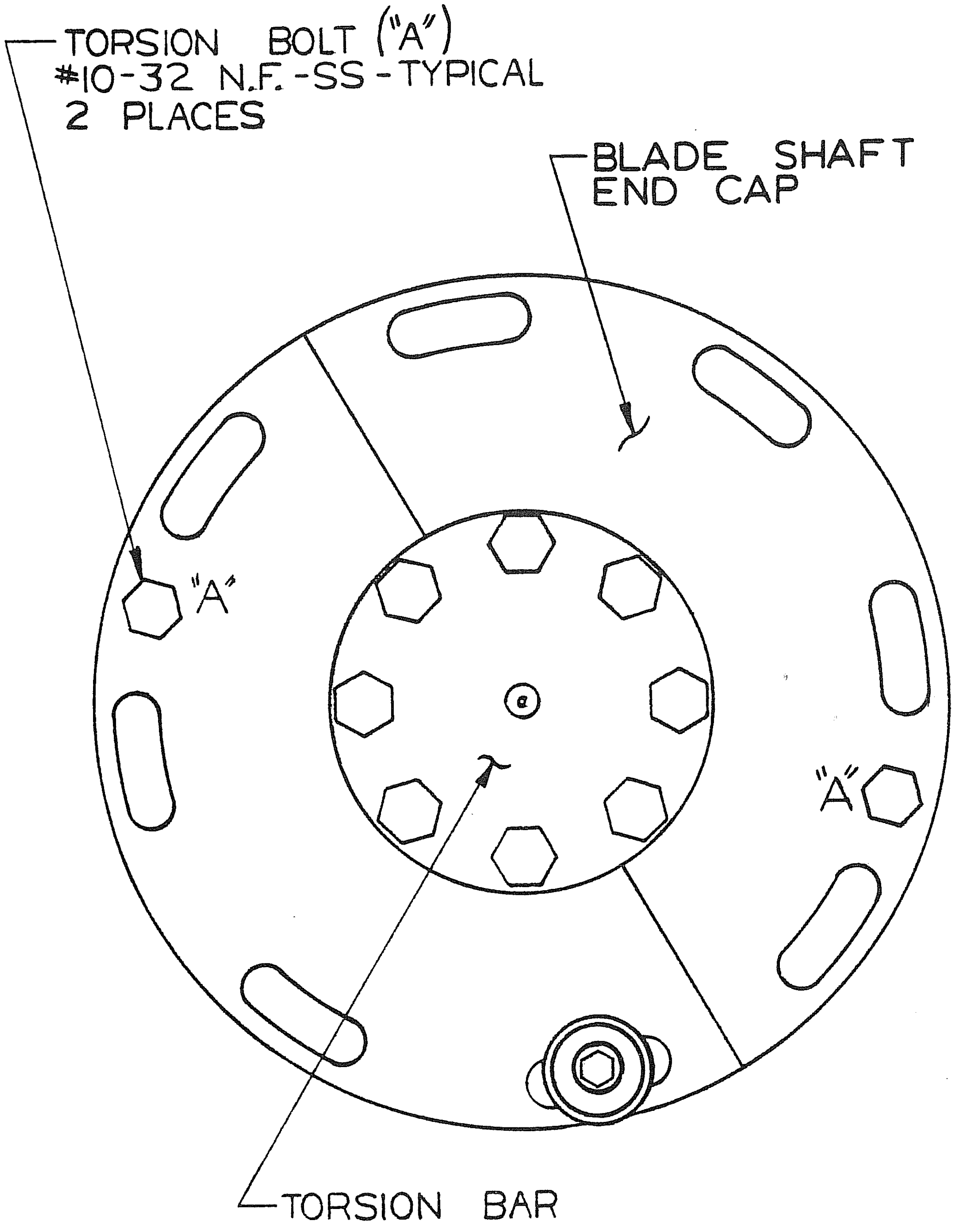


FIGURE 11 : TORSIONAL LOCKING MECHANISM FOR BLADE PITCH.

The original yaw pivot assembly consisted of a yaw pintle shaft, two teflon sleeve bushings, a teflon thrust bearing, and a steel support housing. To reduce the yawing friction, the teflon bearings were replaced with two heavy-duty roller bearings and a needle thrust bearing. The steel housing was redesigned to support the new bearings and to include provisions for lubrication of the bearings. With the yaw breakaway torque reduced to 5 foot-pounds, the wind machine yawed smoothly in winds less than 3.5 m/s during free-yaw tests at Mount Washington. The positions of the brake support and of the slip ring support were changed for greater ease in assembly. The lightning rod bracket was deleted from the design.

3.4 The Drive Train

During component inspection following testing in April of 1979, rust was observed on the brake disk. The material specification for the brake disk was changed from a 1010 steel disk to an austenitic stainless steel disk. The components of the brake actuator mechanism were designed in detail.

The gearbox and alternator were modified slightly to reduce the breakaway torque. The shielded bearing in the gearbox was replaced with an open bearing. In the alternator, sealed bearings were replaced with open bearings and the shaft seals were removed from the unit. These changes reduced the break-away torque of the drive train (driven from the low-speed gearbox shaft) from 40 inch-pounds to 19 inch-pounds.

3.5 The Electrical System

The lightning protection system was re-evaluated and the lightning rod was deleted from the design. Surge protectors in the control box, metal oxide varistors on the slip rings, a grounded frame, and grounded tower and guy wire anchors should protect the wind machine from high voltage surges. The removal of the lightning rod from the design, however, probably does represent a decrease in the protection from direct lightning strikes. The decision to make this change was based on practical problems experienced during testing and on uncertainties about the actual ability of the lightning rod to protect the machine from direct strikes. Among the practical problems was the fact that the lightning rod had to be taken down to open the nacelle for maintenance. This was an inconvenience and presented a potential safety hazard to maintenance personnel if the lightning rod were to be dropped while being handled. In addition, the lightning rod (cantilevered above the top of the wind machine) suffered from vibration problems which could not easily be resolved.

The material for the slip ring case was changed from an opaque plastic to a clear plastic. The purpose of this change was to make it possible to visually inspect the slip rings without removing the case.

3.6 The Fasteners

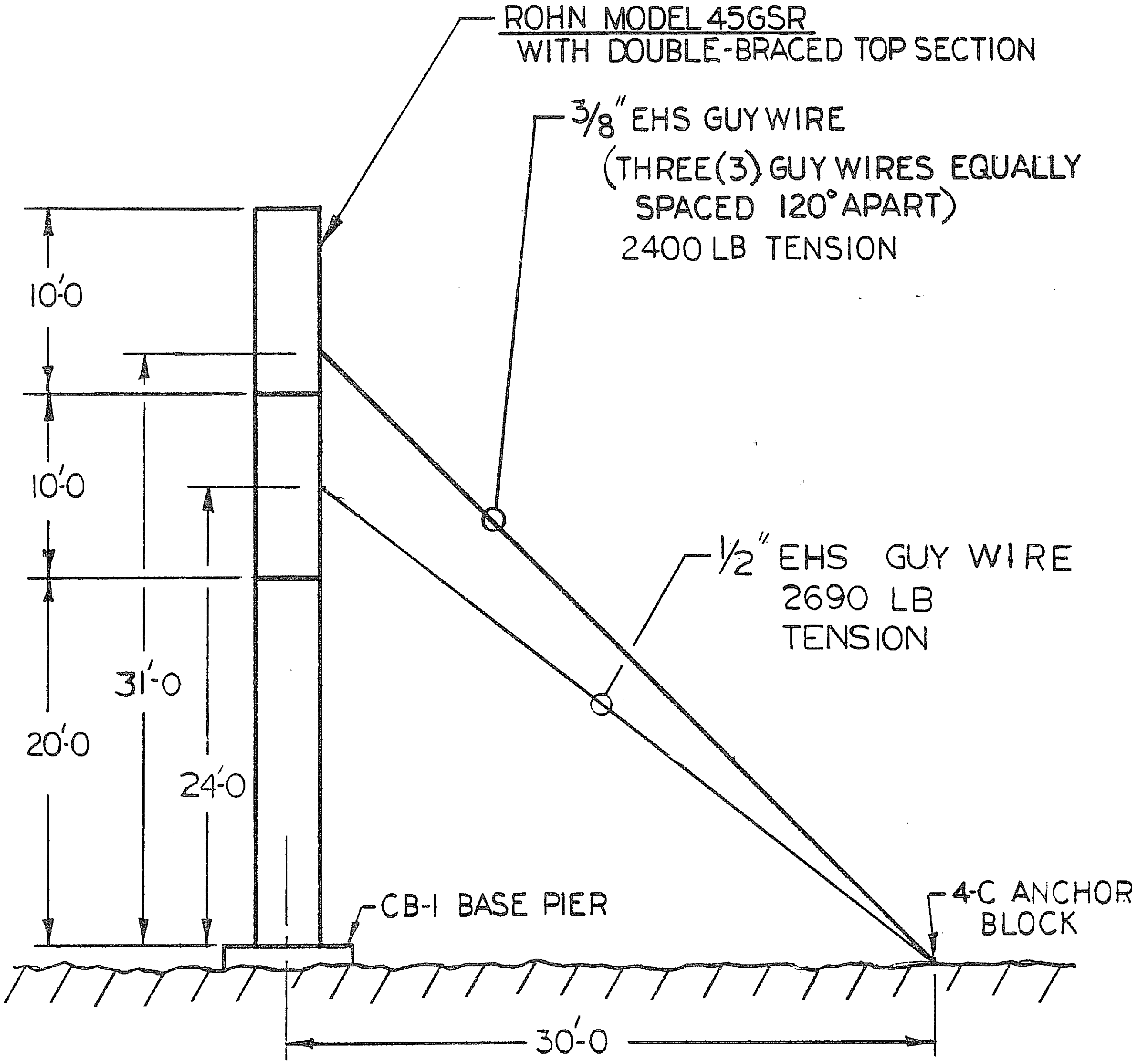
During the final shake-down test at Mount Washington, several fasteners loosened, causing the failure of major components. As a result of these tests, all combinations of lock washers and standard nuts on the machine were replaced with flat washers and lock nuts. All screws were secured by lock washers and Loc-Tite.

3.7 The Tower

Tower/wind machine vibrations occurring at Rocky Flats during buyer testing necessitated the redesign of the tower. A second set of guy cables was added to the tower as shown in Figure 12. The addition of the second set of guy cables changed the predicted values of the tower natural frequencies. For bending of the tower with motion in the direction of the rotor axis, the two lowest frequencies changed from 3.12 and 11.7 hertz to 3.36 and 15.7 hertz. The predictions for the natural frequencies of the redesigned tower are shown in Table 1.

FIGURE 12

TOWER AND FOUNDATION SPECIFICATIONS



FOUNDATION: ROHN INSTALLATION DRAWING # 45GSRSB
MAT'L LIST: ROHN 45GSR TOWER ASSEMBLY FOR ENERTECH

Table 1: Predicted Tower Frequencies

Mode	Frequency
1st Tower Bending	3.36 hertz
1st Bottom Guy Wire	5.32
1st Top Guy Wire	6.16
2nd Bottom Guy Wire	10.64
2nd Top Guy Wire	12.33
2nd Tower Bending	15.70

The Campbell Diagram for the redesigned tower and blades is shown in Figure 13. The new design offers some improvement over the original design by moving the crossing of two-per-revolution excitation with the second tower bending natural frequency further outside the operating range. However, there are still a number of crossings of possible excitation frequencies with system natural frequencies within, or near, the operating range. The crossing of the first tower bending frequency with one-per-revolution excitation near 200 rpm has been identified as a problem. The other crossings in or near the operating range could represent potential vibration problems. Testing of the wind machine on the specified tower should determine which frequencies produce problems. Further design work on the tower is likely needed to eliminate all vibration problems.

3.8 Summary of Design Changes During Phase II

Figure 14 summarizes the design changes occurring during Phase II. Further changes are needed to reduce manufacturing costs and to reduce machine vibrations on the tower.

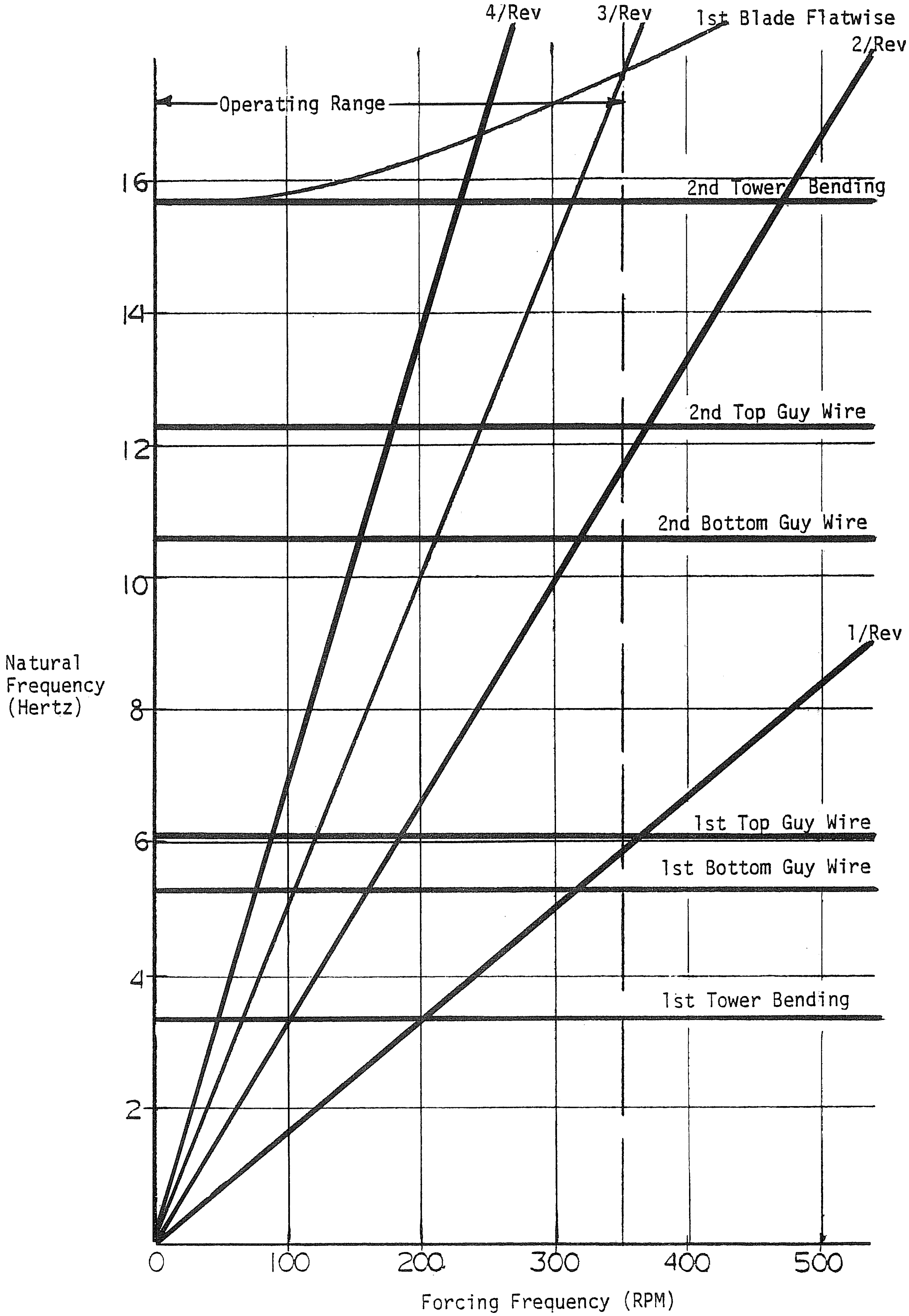


Figure 13. Campbell Diagram

Figure 14: Summary of Design Changes

During Phase II

Design Change	Reason For Change
1. Hub/Rotor Changes	
1.1 Pitch Weights - The length of the pitch weight was increased from 8.1 to 8.98 inches. The weight increases by 1.39 pounds per inch of increased length; thus the increase in weight amounted to 1.22 pounds.	Final truck testing of Prototype 1 showed that an additional 1.25 lbs. was needed on each pitch weight to limit the maximum rotor speed to 350 rpm.
1.2 Torsion Bars - The fillet welds were increased from 3/8 to 5/8 inch.	For blade pitch up to 19^o, the additional weld size was needed for fatigue strength.
1.3 The Blades - The airfoil series was changed from a NACA 23000 series to a flat-bottomed airfoil; the chord at the root was increased from 9" to 10"; the root-to-tip twist was increased to 6^o; the thickness of the blade at the tip was decreased.	A major blade redesign was necessary to improve the rotor start-up characteristics. The effects of this redesign are discussed in Section 6.6.
1.4 Initial Blade Root Pitch - The initial blade root pitch was decreased from 10^o to $9\frac{1}{2}^o$.	The blade root pitch was changed to optimize the performance with the new blades.
1.5 Power Curve - The power curve was revised.	The power curve was revised as a result of contractor testing.
1.6 Hub Welds - The angle of the groove welds in the hub weldment was decreased from 45^o to 30^o.	The welds were changed to decrease the time needed to weld the prototype hubs.
1.7 Blade Back-up Plates - The size of the plates was decreased.	The size was changed to make assembly of the prototypes easier.
1.8 Bearing Tube End Caps - Torsional locking screws were added to the end caps.	The end cap screws "lock" the initial blade pitch.
2. Yaw Pivot Assembly	
2.1 Radial Yaw Bearings - The teflon sleeve bushings were replaced with two heavy-duty roller bearings.	The roller bearings have less friction. The redesign was necessary to reduce yaw break-away torque and allow tracking in light winds.
2.2 Thrust Bearing - The teflon thrust bearing was replaced with a needle bearing.	The change decreased break-away torque and allows tracking in light winds.

Figure 14 Continued

2.3 Steel Housing - The housing was redesigned to support the redesigned bearings. Provisions for lubricating the bearings were included in the design.	The roller bearings required dimension changes in the housing.
2.4 Brake Support - The position of the support was changed.	Position change for ease in assembly.
2.5 Slip Ring Support - The position of the supports changed.	Position change for ease in assembly.
2.6 Lightning Rod Bracket - It was deleted from the design.	The lightning rod was deleted.
3. The Drive Train	
3.1 Brake Disk - The material was changed from 1010 steel to an austenitic stainless steel.	A material change was necessary to prevent rusting of the disk.
3.2 Brake Actuator Mechanism - The components were designed in detail.	Design improvements were necessary to reduce the assembly time.
3.3 Gearbox - A shielded bearing was replaced by an open bearing.	The change reduced break-away friction.
3.4 Alternator - Sealed bearings replaced open bearings. The shaft seals were removed.	The change reduced break-away torque.
4. The Electrical System	
4.1 Lightning Rod - Was deleted from design.	The removal of the lightning rod is discussed in Section 3.5.
4.2 Slip Ring Case - Changed from a gray plastic case to a clear plastic case.	Change reduced assembly time, allowed easier inspection of slip rings.
5. Fastener System	
5.1 Bolts, Nuts, Washers - The lock washer/standard nut combination was changed to a flat washer/lock nut combination.	Several fasteners came loose during Mt. Washington test, the revised locking mechanisms should not loosen as easily during vibrations.
6. Tower - A second set of guy wires was added to the tower.	Machine/tower interactions occurred during atmospheric testing at Rocky Flats. The guy cables were added to stiffen the tower and improve the vibration characteristics.

4. Reliability and Failure Rate Analysis

In this analysis, the overall system was divided into subcomponents and the failure rate values for each subcomponent were determined. The design goal for reliability of the 2kW wind machine was a mean time between failures (MTBF) of at least 10 years.

For the purpose of the reliability analysis, the system was divided into the following component/subcomponent listings and numbering system:

1.0 Rotor Assembly
- 1.1 Blades
- 1.2 Blade holder and shaft assembly
- 1.3 Blade shaft bushing
- 1.4 Tension/Torsion bar assembly
- 1.5 Main hub weldment

2.0 Gearbox Assembly
- 2.1 Gearbox
- 2.2 Gearbox/Alternator coupling assembly

3.0 Alternator Assembly

4.0 Slip Ring Assembly

5.0 Main Frame Assembly

6.0 Yaw Pivot Assembly
- 6.1 Yaw pivot weldment
- 6.2 Yaw bearing

7.0 Tower Structure
- 7.1 Tower
- 7.2 Tower foundations

8.0 Lead-In Wiring
- 8.1 Wire couplings
- 8.2 Lead-in wire

9.0 Control Panel
- 9.1 Main power fuses
- 9.2 Terminal board

Only the components which were considered critical to the proper operation of the system were included in the reliability analysis. Figure 15 shows the reliability block diagrams for the system. Figure 15A shows that, if appropriate failure rate values are computed for each assembly, the failure rate for the overall system can be found by summing these values. Figure 15B shows how the failure rate values for each assembly were obtained. The failure rate values for each assembly were determined by adding up the failure rates of the individual components in series within the assembly.

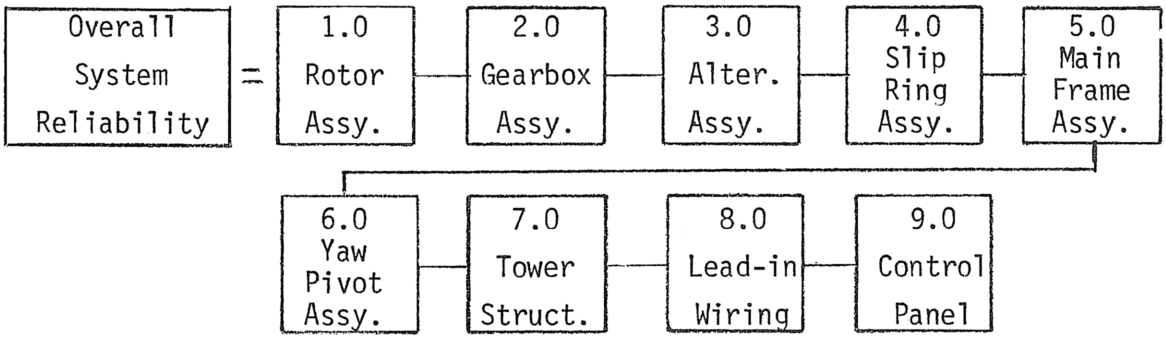
In some cases a corrected failure rate value was used in order to obtain a more realistic overall failure rate. Merely adding up all the individual subcomponent values to obtain a total for the system would give a correct value only if every subcomponent failure resulted in a system failure. Careful examination of the effects on the system of each subcomponent failure shows that this is not really the case. Because of this a correction factor has been introduced which appears as a multiplier in Table 2. This multiplier takes into account the percentage of failures of a given subcomponent which might cause complete system failure.

Table 2 shows a tabulation of the failure rate values and correction multipliers for each subcomponent. Also indicated is the source of each failure rate value. Most values were estimated from tables of generic failure rate data.

The reliability data for the alternator assembly was supplied by Maremont Corporation, the manufacturer of the alternator. This data is contained in Maremont's report ALTMS-1, "Reliability Data for Enertech Windmill System Electrodyne Alternator E200-24," dated August 10, 1978. The reliability data was based on Maremont warranty return information and vendor data. Because the Maremont report gives the calculated reliability for the alternator assembly, the reliabilities of the individual subcomponents of the alternator were not considered in this analysis. The alternator reliability for wind machine applications was calculated to be .9971 for one year and .942 for ten years. Based on the formula $R=e^{-\lambda t}$ (where R is reliability, λ is failure rate, and t is time), the resulting

Figure 15: Reliability Block Diagrams

A. System Reliability Block Diagram



B. Component Reliability Block Diagrams

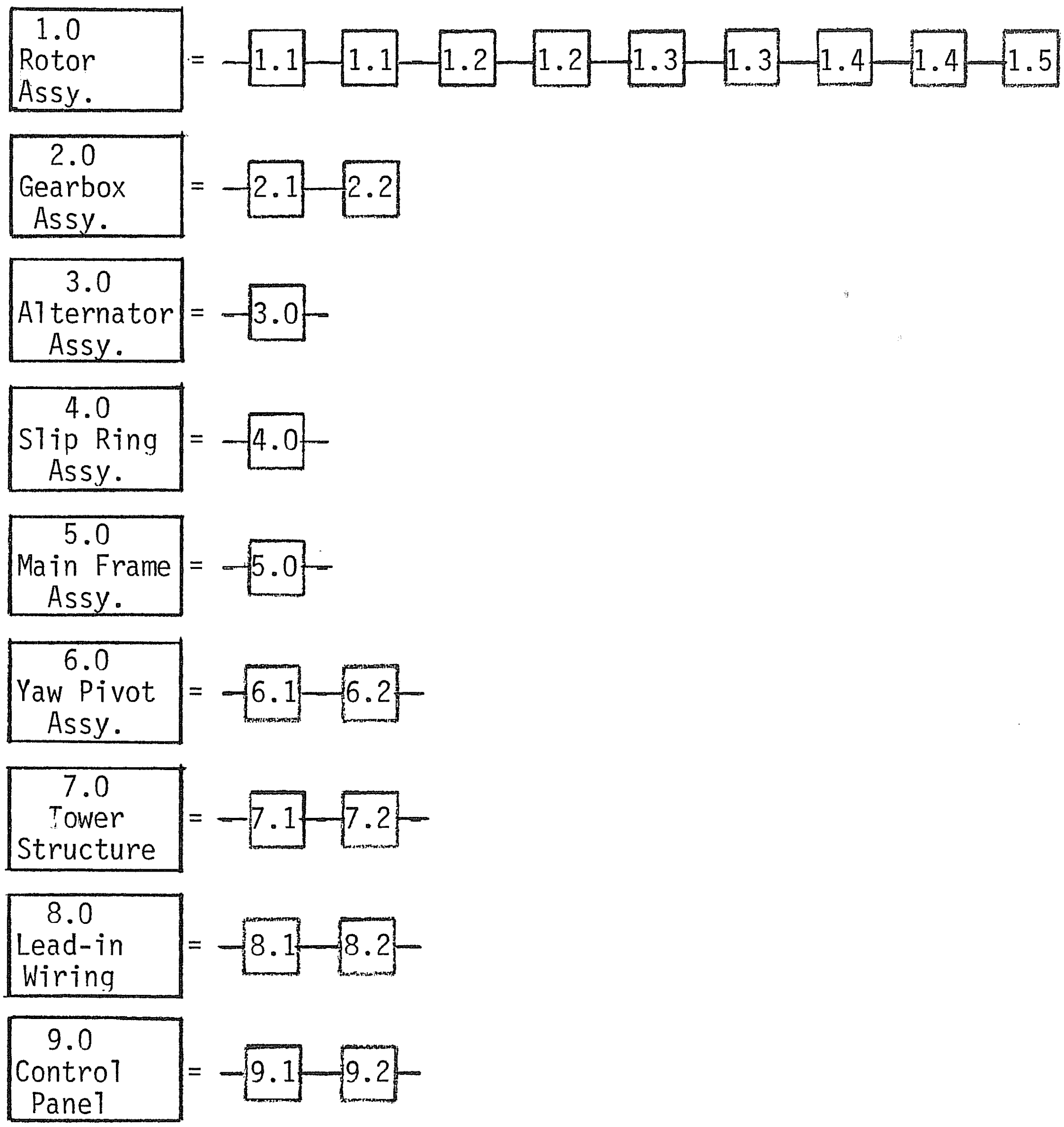


TABLE 2: Failure Rate Values and Multipliers for Individual Subcomponents

Part No.	Description	(1) λ	(2) Source	Qty. per Machine	Multi-plier	(3) λ_c
1.1	Blade	.10	E	2	1	.20
1.2	Blade holder and and shaft assembly	.05	E	2	1	.10
1.3	Blade shaft bushing	.50	E	4	.5	1.0
1.4	Tension/torsion bar assembly	.055	E	2	1	.11
1.5	Main hub weldment	.05	E	1	1	.05
2.1	Gearbox	3.0	E2	1	.9	2.7
2.2	Gearbox/alternator coupling assembly	.7	E1	1	.9	.63
3.0	Alternator assembly	.682	M	1	1	.682
4.0	Slip ring assembly	2.0	E3	1	.9	1.8
5.0	Main frame assembly	.01	E1	1	1	.01
6.1	Yaw pivot weldment	.01	E1	1	1	.01
6.2	Yaw bearing	.50	E	2	.5	.50
7.1	Tower	.02	E	1	1	.02
7.2	Tower foundations	.01	E	1	1	.01
8.1	Wire couplings	.05	E3	1	1	.05
8.2	Lead-in wire	.03	E3	1	1	.03
9.1	Main power fuses	.5	E	1	1	.5
9.2	Terminal board	.05	E3	1	1	.05

TABLE 2 Continued

Notes: (1) λ= Failure rate in number of failures per million hours of operation.

(2) Sources for failure rate values listed are as follows:

E = Estimated failure rate based on best available generic data.

E1= Mean failure rate value from Table 18.3, Generic Failure - Rate Distributions, Mechanical Design and Systems Handbook, Harold Rothbart, ed., McGraw-Hill, 1964.

E2= Failure rate for two-stage helical gear train assemblies from NACA Report #TR-824, available from Superintendent of Documents.

E3= Failure rate value from Navships 93820, Reliability Prediction Handbook.

M = Failure rate based on Maremont data.

(3) λ_c= Failure rate for subcomponent corrected to reflect effect of subcomponent failure on overall system failure rate. (λx Qty. Per Machine x Multiplier)

TABLE 3 Summary of Corrected Failure Rate Values for Components

Part No.	Description	Failure Rate (failures/10^6 hr.)
1.0	Rotor Assembly	1.46
2.0	Gearbox Assembly	3.33
3.0	Alternator Assembly	0.68
4.0	Slip Ring Assembly	1.80
5.0	Main Frame Assembly	0.01
6.0	Yaw Pivot Assembly	0.51
7.0	Tower Structure	0.03
8.0	Lead-in Wiring	0.08
9.0	Control Panel	0.55
	Complete system ($\Sigma\lambda_c$)	8.45

MTBF = 118,000 hours = 13.5 years

failure rates are:

λ =.332 x 10^{-6}/hr. (based on one-year reliability)

λ =.682 x 10^{-6}/hr. (based on ten-year reliability).

For the analysis, the more conservative of these two values (λ=.682 x 10^{-6}/hr.) is used.

Referring to Figure 15 and Table 2, it is possible to summarize the overall system reliability. This is done in Table 3. From Table 3, it can be seen that the failure rate for the overall system adds up to 8.45 failures per million hours of operation. If 24 hour-per-day operation is assumed, this results in a mean time between failures of 118,000 hours or 13.5 years. Thus, the design exceeds the minimum design specification of ten years MTBF.

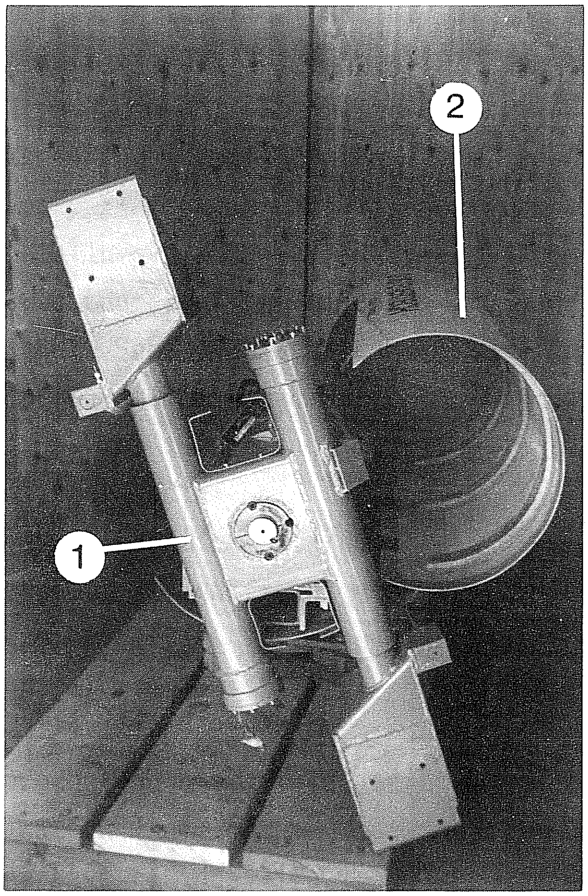
5. Fabrication of the Wind Machine

The fabrication of the 2kW wind machine consists largely of the assembly of purchased components. The hub, the blades, the supporting frame, and the nacelle were fabricated by contractors according to Enertech's specifications. The gearbox, the slip rings, and the components for the control box are all purchased parts that require modifications. The alternator, the tower, the bearings, and seals are purchased components requiring only assembly. The brake actuator mechanism and the nacelle brackets were fabricated at Enertech from steel structural shapes. The slip ring case was fabricated from plastic pipe and pipe caps.

A rotor hub for the 2kW wind machine is shown in Figure 16. The hub weldment, blade shafts, torsion bars, and pitch weights were fabricated from steel tubing, bars, and plates. Following the machining and welding of the blade shaft assemblies, the bearing surfaces were hard-chrome plated and then polished.

The torsion bars, shown in Figure 17, were fabricated from stainless steel bar as follows: 1) The bar was cut to length and the bar end caps were cut and drilled. 2) The caps were welded to the bar. 3) The bar was ground and polished. The welds were ground to a smooth fillet; the bar was polished to bring the diameter to the desired value and to obtain the desired surface finish. 4) The bar was heat treated.

The fabricator reported problems grinding the bar to its specified diameter. Because of its slenderness, vibration problems were encountered during machining.



1. Hub
2. Nacelle
3. Nacelle
4. Slip Rings
5. Alternator
6. Hub
7. Gearbox

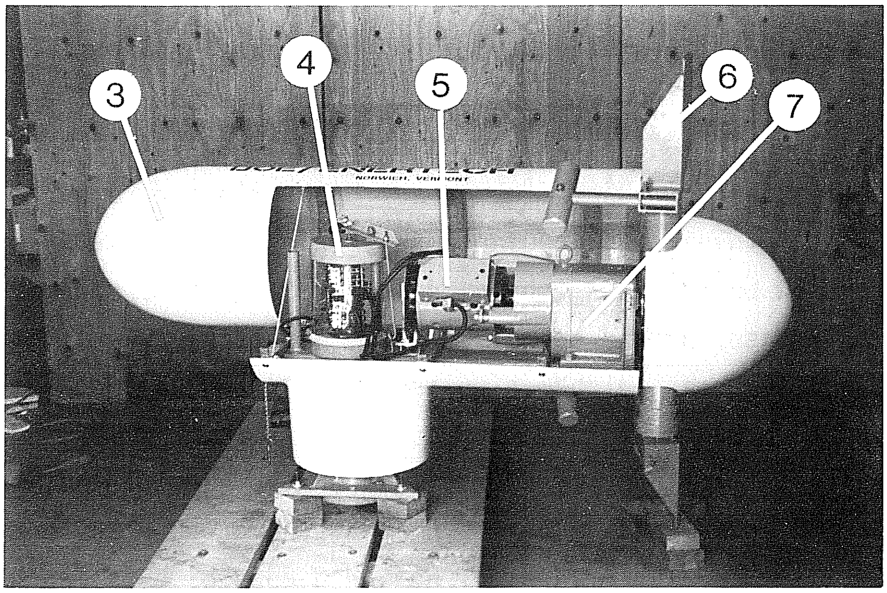


FIGURE 16: ASSEMBLED ROTOR AND ASSEMBLED DRIVE TRAIN

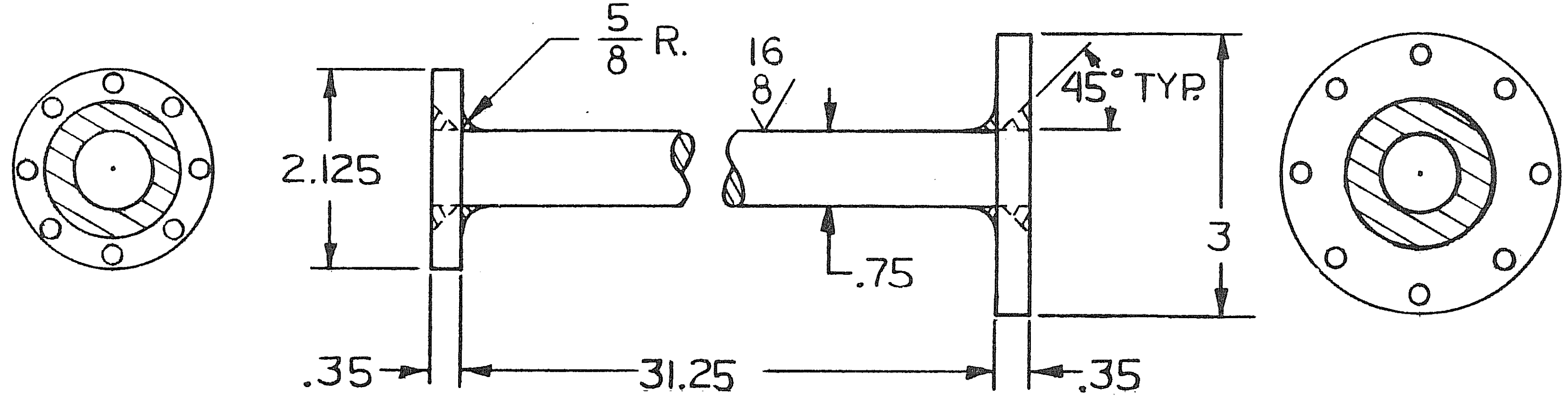


FIGURE 1 7 : SKETCH OF TORSION BAR

Problems with the final quality of the welds were also encountered. After summer truck testing in 1979 the bars were inspected for signs of fatigue damage. Following a discovery of what were believed to be cracks in the throats of the fillet welds, one of the bars was sent to the Rocky Flats test facility for further analysis. After analysis at the metallurgy laboratories, several conclusions were reached. First, the presumed cracks were joints where welds specified by Enertech were not made. Second, the welds present contained many defects: lack of full penetration, hot cracks, and porosity. Third, the weld filler metal was not the optimum material for the base metal. Fourth, "the microstructures of both the bar and the flange showed a high concentration of chromium-rich/nickel-lean stringers, possibly ferrite. The presence of a significant amount of this second phase may result in fatigue properties that are different from those that were used for design calculations."

For production units in quantities greater than 100, this method of fabrication would not be recommended. To reduce production cost and to increase fatigue life, the bars would be forged in one piece. However for quantities of less than 100, the present method of welding the torsion bars would be used.

The blades were fabricated "by hand" by Peter Baldwin of Brooks, Maine. After the sitka spruce blanks were laminated together, the desired airfoil shape was cut into the blade at several places along its length. The blades were then planed to the airfoil shape between cuts and sanded. Following receipt of the blades at Enertech, the airfoil shape was checked for conformance to templates and the blades were coated with an epoxy sealant and then painted. For production units (in quantities over 100) the blades would probably be shaped on a machine, sanded, and painted.

Following the assembly of the hub components, the initial blade pitch was set and the hub was balanced on a tire balance. The two blades were then balanced to each other on the hub.

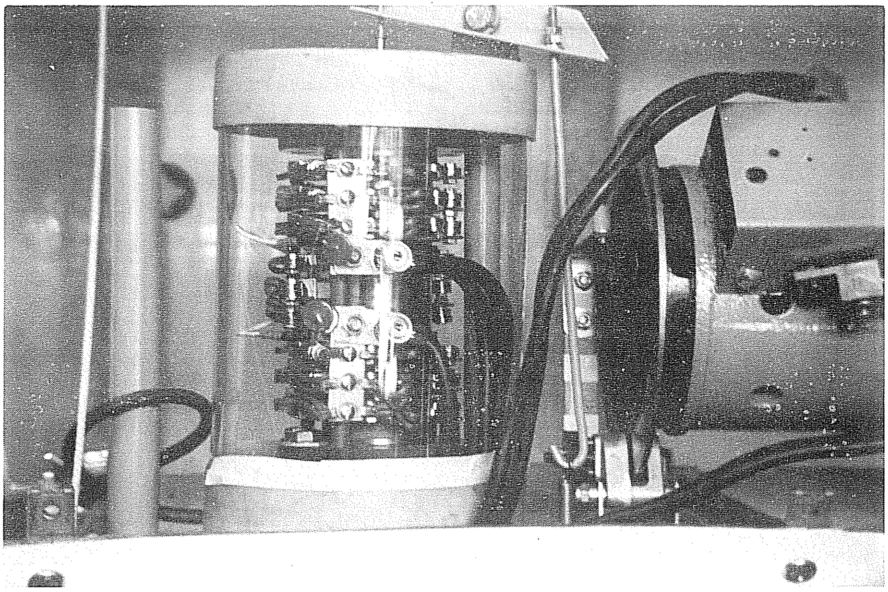
The drive train components were purchased from other manufacturers. The alternator, developed for this wind machine by Maremont Corporation, is currently in production. The gearbox is a purchased component with a modified low-speed shaft. The maintenance brake is also a purchased component. The assembled drive train is shown in Figure 16.

The fabrication of the supporting frame consists of the assembly of both off-the-shelf components and fabricated components. The supporting frame, shown in Figure 10, consists of the main frame/yaw pintle weldment, the slip ring tube weldment, the yaw brace weldment, the two radial bearings, the needle thrust bearing, and the bearing seals. The steel supporting structures were made from structural tubing, shapes, and plate. No unusual problems were encountered with the fabrication or the assembly of these components.

The fabrication of the electrical components consists of the assembly of purchased components which require minor modifications. The slip rings, shown in Figure 18, require modification of the end flanges. The control box, shown in Figure 19, also consists of an assembly of purchased components.

The fiberglass nacelle was fabricated by Ocean Research of Warwick, Rhode Island. Following a layer of white gel coat, alternating layers of resin and fiberglass material were applied in a female mold. Balsa wood ribs were sandwiched between the last two layers of material to stiffen the nacelle. Following the receipt of the nacelle at Enertech, hinges and brackets were fitted to the nacelle. A finished nacelle is shown in Figure 18.

The recommended tower, shown in Figure 12, is purchased from Unarco Rohn and assembled at the wind site. Guy wire tensions were measured either by a tension meter or by measuring the guy wire frequencies. The natural frequency of the top guy wires should be 6.16 cycles/second and the natural frequency of the bottom guy wires should be 5.32 cycles/second.



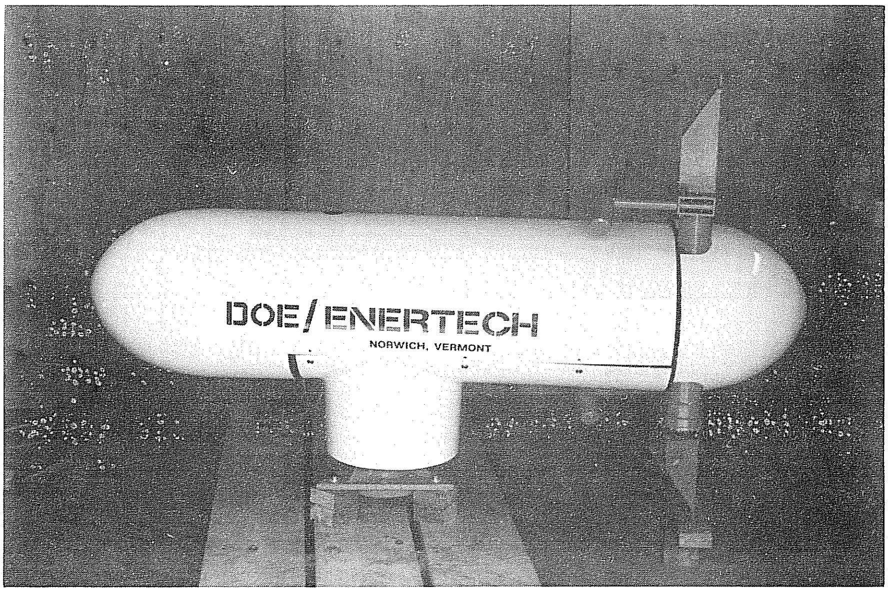
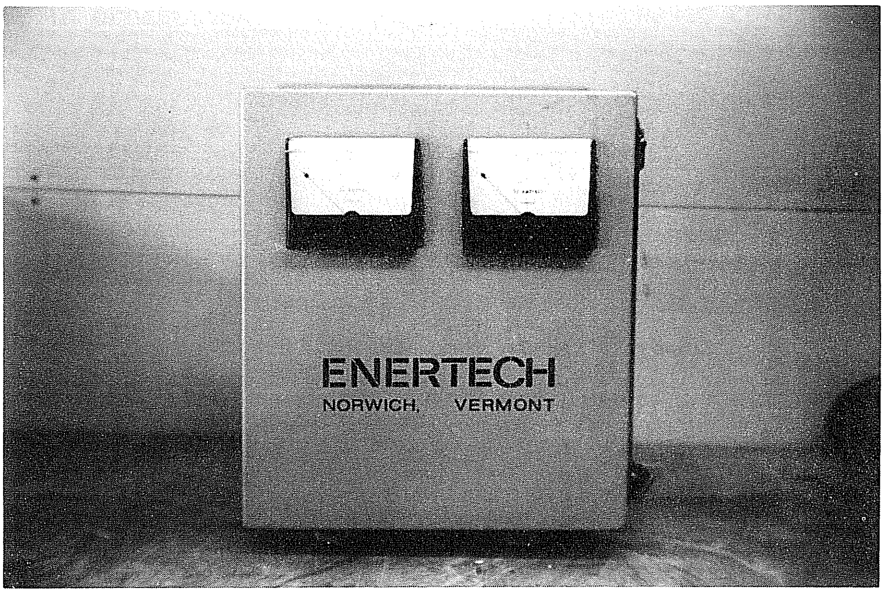


FIGURE 18: ASSEMBLED SLIP RINGS AND ASSEMBLED NACELLE



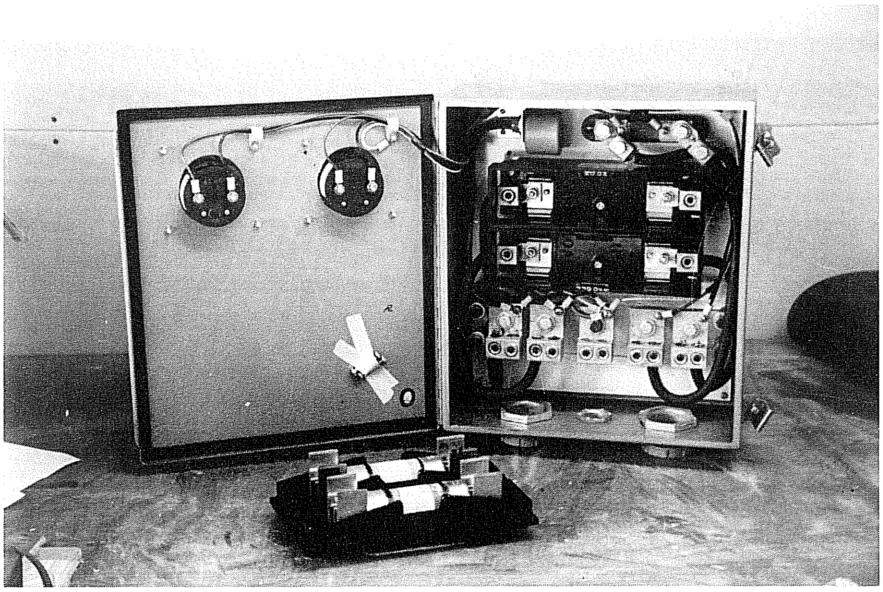


FIGURE 19: 2KW CONTROL BOX

6. Summary of Contractor Testing

6.1 Introduction

The purposes of the Phase II contractor test program were to verify that the design met the contract specifications, to observe the operation of the prototype wind machine and record its performance, and to find and correct as many problems with the wind machine as possible while it was at the contractor's facility. The Phase II test program consisted of both atmospheric tests and controlled-velocity truck tests of the prototype unit. During the test program, the performance of the machine was recorded and was found to meet the required power output specification. The pitching rotor was shown to be effective in controlling the maximum rotor speed. A number of problems with the design were also discovered and corrected during the testing. However, because of the limited testing time, the machine was not completely debugged during the contractor test program.

After testing was completed at Enertech, the prototypes were sent to Rocky Flats for buyer testing. During these tests, several more problems were uncovered, most notably the poor start-up characteristics and vibrations of the wind machine and tower. One prototype was returned to Enertech and, in October of 1980, an additional program was begun under a fixed-price contract to improve the start-up of this prototype. At the completion of this program, the wind machine would start in 4.5 m/s winds. The machine was then returned to Rocky Flats for additional testing.

The following sections describe the testing of the wind machine done by the contractor during Phase II.

6.2 Enertech Field Tests of Prototype 1

The 2kW wind machine was installed on a tower at the Enertech test site in Norwich, Vermont for atmospheric testing. The machine that was tested consisted of the components of Prototype 1 that were available at the time of testing with the remaining components coming from the Phase I test machine. The frame from Prototype 1 was combined with the alternator and rotor from the Phase I test unit.

The objectives of the test were to observe the operation of the machine in a free-yawing condition, to measure performance under atmospheric conditions, to observe the vibration characteristics of the machine on the specified tower, and to check for any problems in installing the complete unit on the tower.

The test facility was set up in March of 1979. The 40-foot Rohn 45GSR tower was erected on a concrete foundation. The tower was guyed 31 feet above its base with three 3/8-inch diameter extra-high-strength guy wires. The test facility also included an electrical load and the wiring between the wind machine and the load. The load consisted of a set of batteries to be charged by the machine, plus a resistance load.

The machine was installed on the tower on April 11, 1979. The machine was instrumented to measure the alternator rotational speed and the output voltage and current. The alternator speed was measured with a magnetic pickup and a gear mounted to the alternator shaft. The output current was measured with a current shunt and ammeter. The voltage was displayed on an analog voltmeter. To measure the wind speed, an R. M. Young anemometer was mounted to the tower at a level below the rotor.

The machine remained on the tower until early May but no significant power output was observed during this test. The lack of power output was partly explained by the fact that there were no substantial winds during the test period. In addition, two problems with the wind machine prevented the machine from responding to what winds there were. The first problem was that the machine would not yaw in light winds because of excessive friction in the yaw pivot. The second problem was that the brake was dragging due to run-out of the brake disk and to rust which formed on the steel disk during the test. After the April tests, the yaw pivot assembly was redesigned using roller bearings instead of teflon bushings. The brake disk was specified to be made from stainless steel and to be machined flat after fabrication.

The complete machine, except for the lightning rod, was installed on the tower for a short period in June of 1979 as a demonstration installation for personnel from the Rocky Flats test center. The machine incorporated the new brake disk and the new yaw pivot assembly which allowed the machine to yaw in light winds. During the time the machine was on the tower, the winds were again light and no output was observed.

The results from the tower testing were minimal. The machine assembly went together on the tower with only minor difficulties. The lack of response to light winds led to new designs for the yaw pivot and brake disk. At the conclusion of this test, there still had been no observation of operation of the machine on the specified tower, of operation while freely yawing, or of performance in atmospheric winds.

6.3 Final Truck Tests on Prototype 1

During June of 1979, final controlled-velocity tests of Prototype 1 were conducted by the contractor. The results of the final tests were as follows: 1) At the rated wind speed of 9 m/s, output power averaged 2.1 kW; 2) The rotor start-up wind speed was 6.5 m/s; 3) The maximum rotor speeds observed were 335 and 350 rpm for an electrically loaded and an electrically unloaded machine, respectively. The 350 rpm rotor speed corresponds to a 19° total pitch travel and to a wind speed of 12 m/s. Figure 20 shows the power curve for the tests. Except for the high start-up wind speed, the wind machine performed as designed. Figure 21 shows the relationship of rotor speed and wind speed for the tests.

The test truck facility is shown in Figure 22. Test results were recorded by a six-channel Honeywell Visicorder chart recorder. Wind speed, rotor speed, blade pitch, voltage, and current were all recorded. Figure 23 shows a schematic of the test truck instrumentation. Wind speed data was measured at the front of the test truck by an R. M. Young anemometer and rotor speed data was measured at the alternator. Blade pitch was measured by strain gages attached to the torsion bars and the strain signals were transferred through slip rings on the rotor to the recording instrumentation. Figure 24 shows the orientation of the strain gages on the torsion bar. Since the tension/torsion bars operate well below their elastic limit, torsional strain is directly proportional to blade pitch.

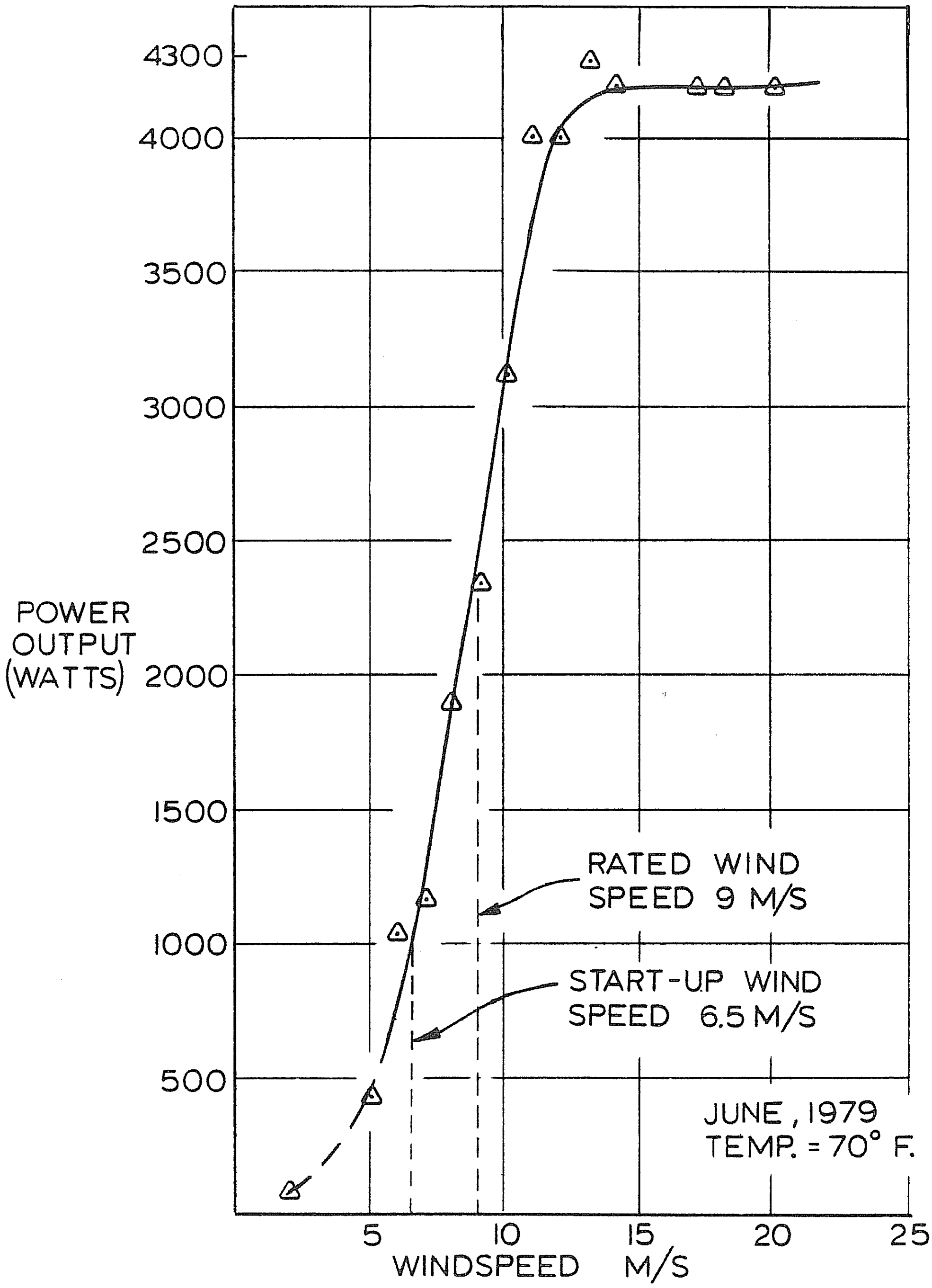


FIGURE 20: POWER VS. WINDSPEED

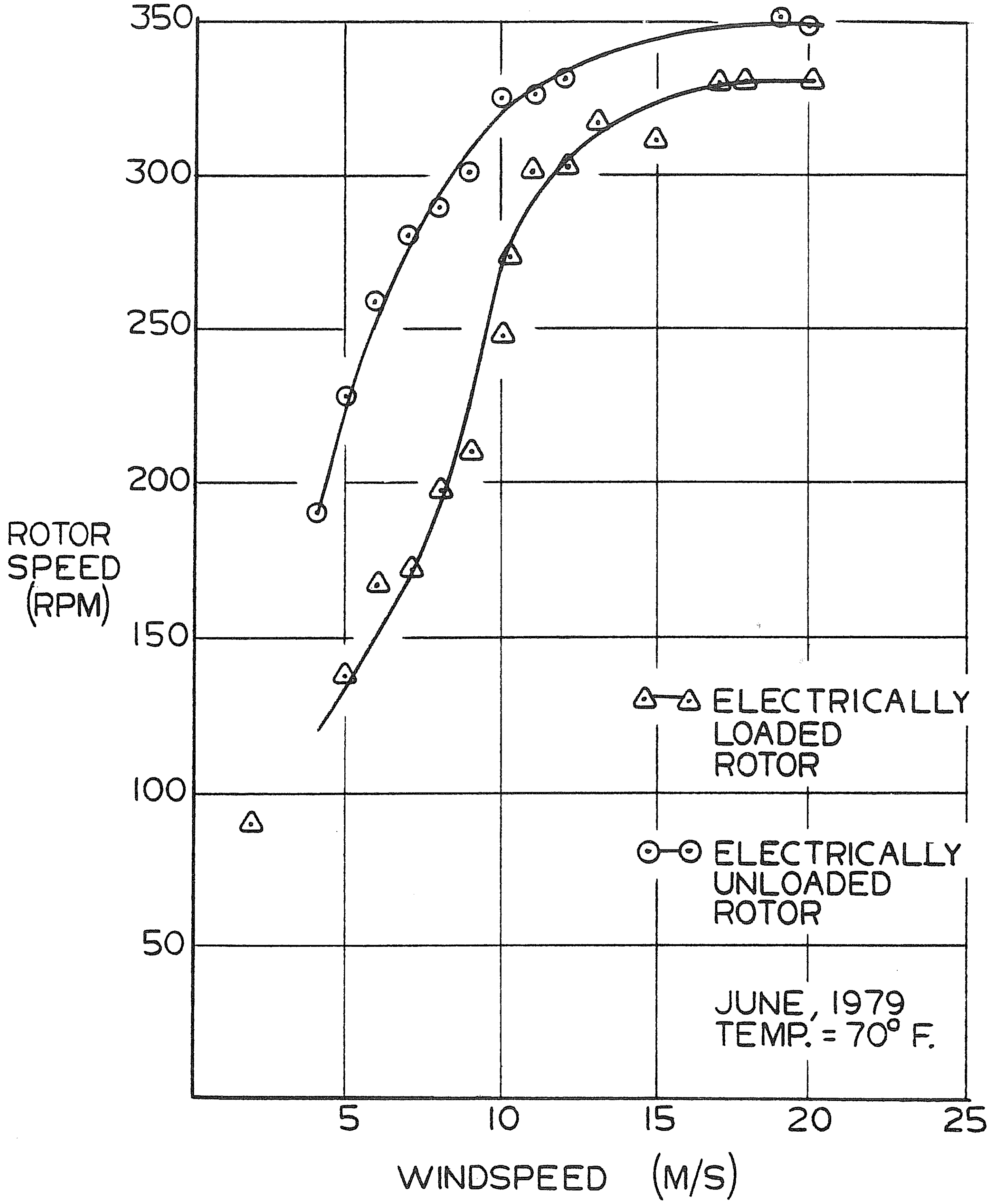
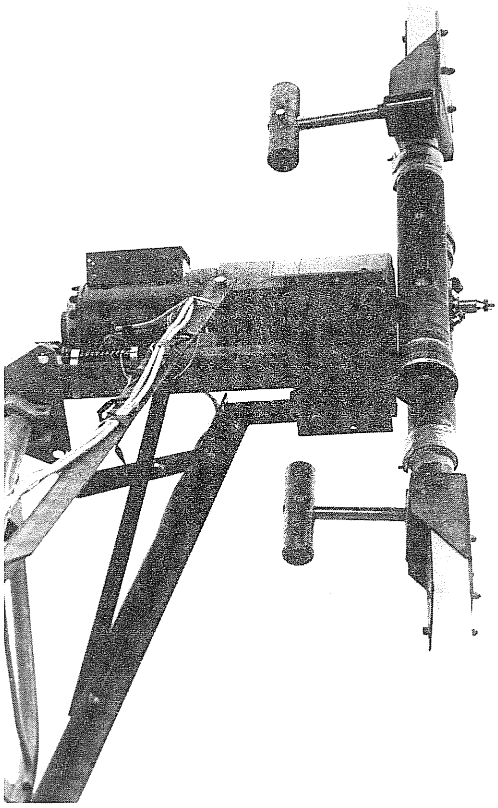


FIGURE 21: ROTOR SPEED VS. WINDSPEED



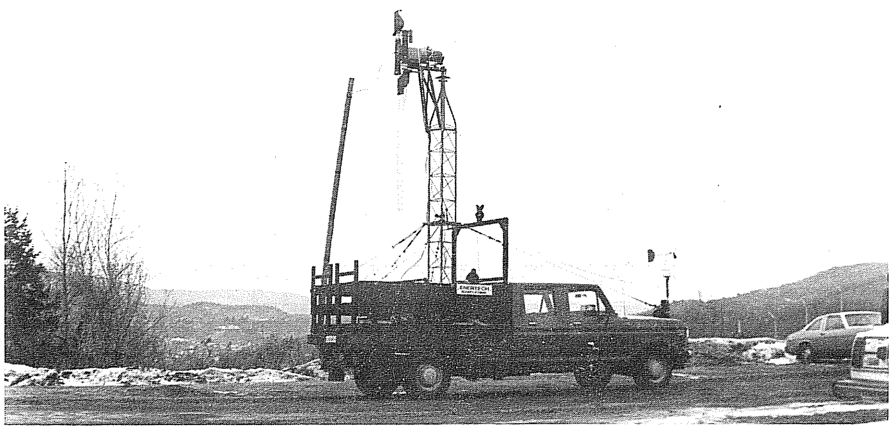


FIGURE 22: TRUCK TEST SET-UP

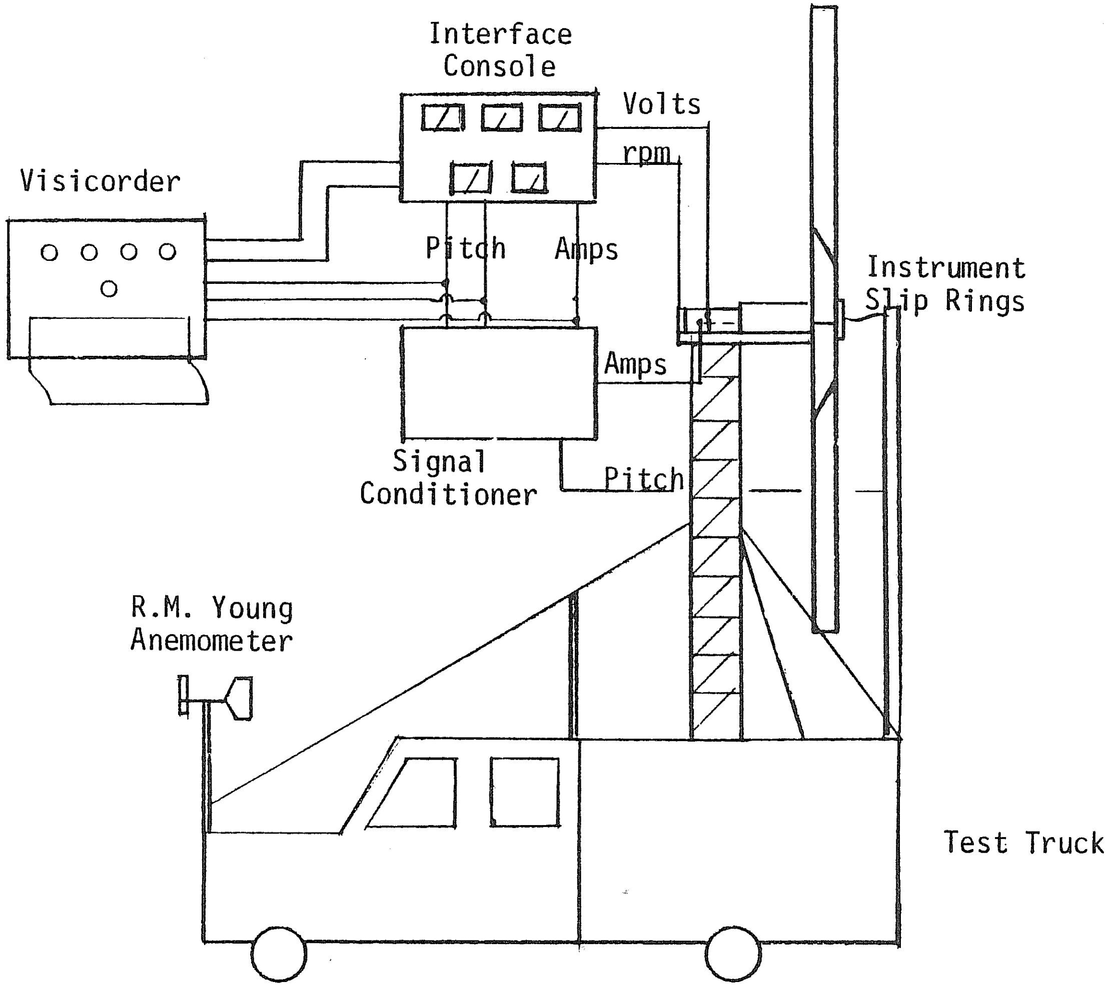
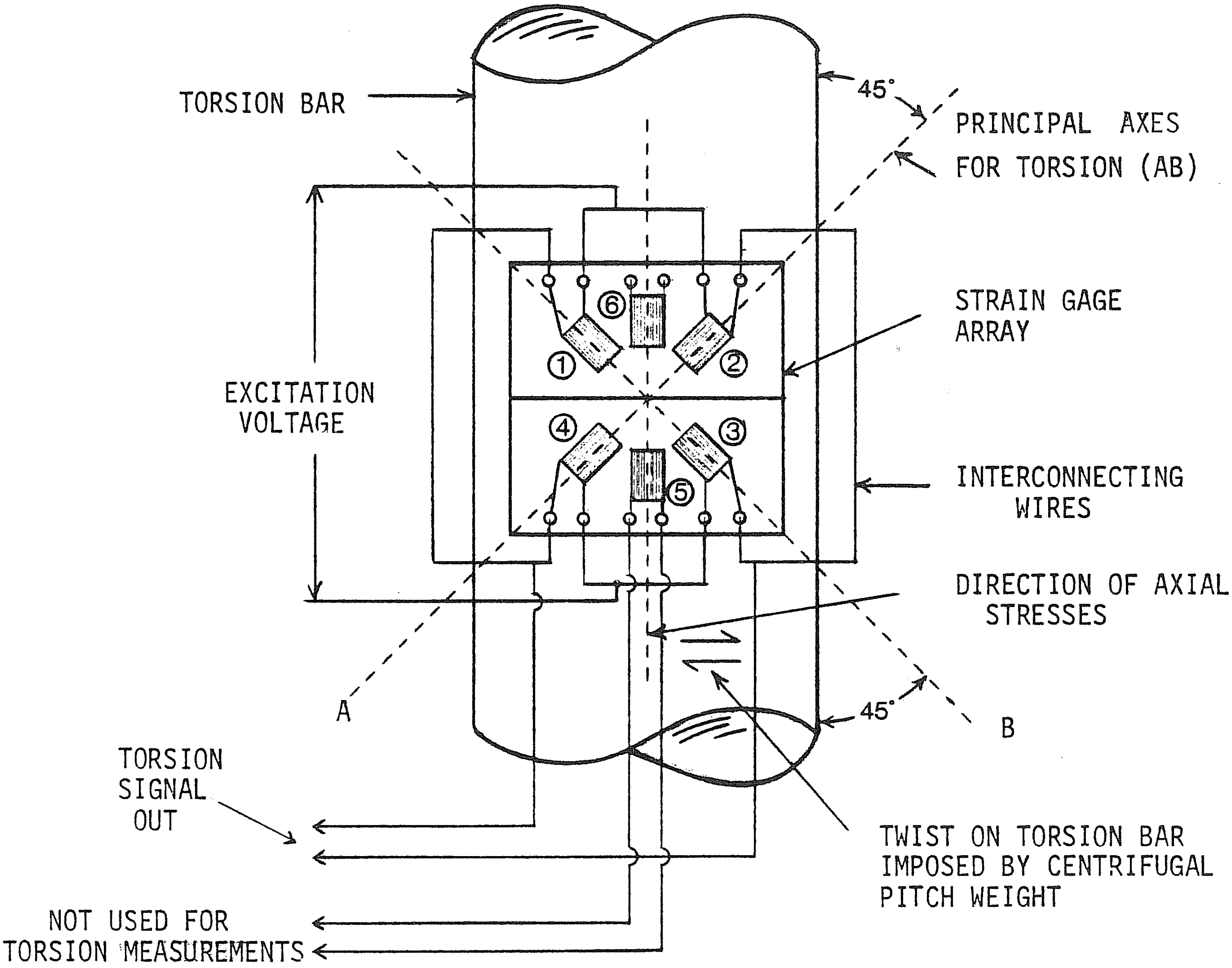


FIGURE 23: SCHEMATIC OF TEST TRUCK

FIGURE 24

STRAIN GAGE INSTALLATION ON TORSION BAR



A console with a metered face was constructed to interface the instrument signals with the Visicorder. A strain gage amplifier was used as a signal conditioner for the current and the blade pitch signals. The outputs from the strain gage amplifier drove the respective recorder inputs directly and scaling factors were set by adjusting the calibrated amplifier gain controls. The console converted the analog wind speed, voltage, and rotor speed signals to the proper voltage and impedance levels for the Visicorder. Each channel had its own gain or scaling control which was easily set using built-in calibration circuitry that simulated various input conditions. In addition, meters on the console displayed blade pitch, voltage, current, and rotor speed for reference during testing. The Visicorder utilizes a special cathode ray device which leaves traces on photosensitive paper. A sample of the output from the Visicorder is shown in Figure 25. After the testing, the data was analyzed and data points were selected where steady-state conditions were recorded. In preparing the performance graphs, the independent variable (for instance, wind speed in a power vs. wind speed graph) was divided into intervals and the data points falling into each interval were averaged.

During the tests early in June, problems with the wind machine were encountered. In the earlier part of the month, 20° of pitch travel was required to stall the rotor at 380 rpm. This imposed higher stresses than those from the 18° and 360 rpm design condition. Static tests on the torsion bars indicated that the spring constants of the bars were within 2% of each other but that both spring constants were 10% higher than the design spring constant. To make the rotor limit at a lower speed, additional mass was added to the pitch weights and the initial blade pitch was reduced by one degree. For the final tests, the initial blade root pitch was 9°, the maximum rotor speed was 350 rpm, and the total pitch travel was limited to 19°. Except for the 6.5 m/s wind speed necessary for start-up, the unit performed as required. For field units, it was assumed that, after an initial wear-in period, the torque necessary to start the drive train would be decreased and the unit would start in 4-5 m/s winds.

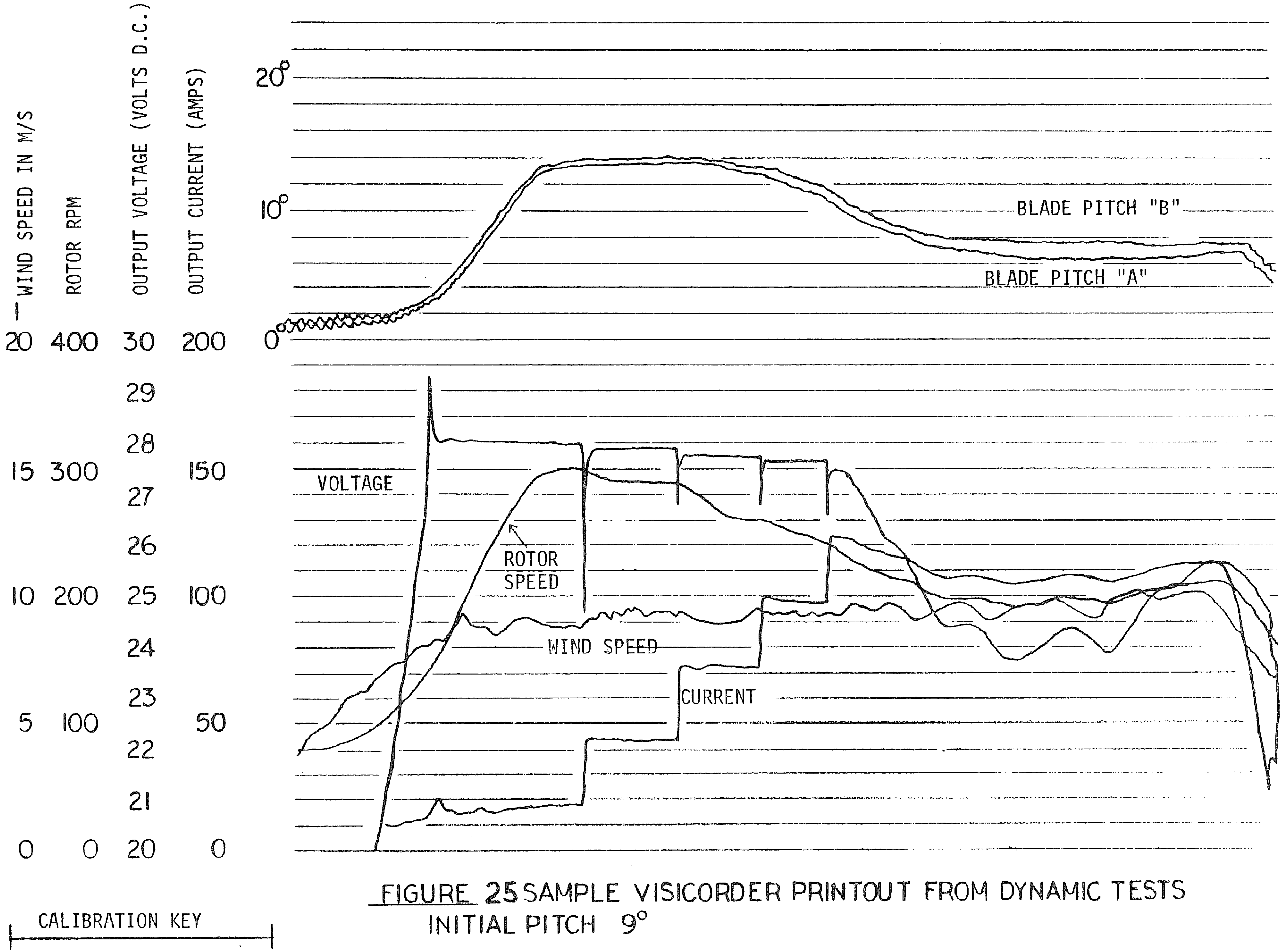


FIGURE 25 SAMPLE VISICORDER PRINTOUT FROM DYNAMIC TESTS
INITIAL PITCH 9°

6.4 Testing the Wind Machine on Mount Washington

In June of 1979, the 2kW wind machine was installed on Mount Washington for final contractor testing. Prior to the Mount Washington tests, the wind machine had not produced power in a freely yawing mode. During truck tests the unit was locked in yaw; during atmospheric tests at Enertech light winds and a stiff yaw pivot assembly precluded power production in a freely yawing mode. The objectives of this test were to observe 1) the down wind tracking characteristics, 2) the vibration characteristics of the wind machine during yawing maneuvers with the rotor spinning, and 3) the overall performance of the wind machine in strong gusty winds. Because the average wind speed at Mount Washington is about 16 m/sec (35mph), the test provided the opportunity for an accelerated atmospheric test.

The test rig, shown in Figure 26, was anchored on a concrete pad just below the summit of the 6288-foot mountain. In addition to guy wires securing the tower to the truck bed, guy wires were attached from the tower (six feet from the top) to eyes in the concrete pad. The three ground mounted cables were tensioned by turnbuckles to approximately 500 pounds each. For additional stability, blocks were placed under the truck frame. The resulting installation was a temporary but rigid structure supporting the wind machine at a hub height of just over 20 feet above the ground.

The instrumentation used during the test consisted of a voltmeter, an ammeter, and a wind speed meter. Because of the time and expense involved in full documentation of test results at this remote site, the test was conducted in a qualitative manner rather than a quantitative manner.

The wind machine was tethered to the Mount Washington test pad from June 29 through July 3. During the seven hours of operation, the winds ranged up to 22 m/s (50 mph) with gusts as high as 35 m/s (78 mph) recorded at the mountain top.

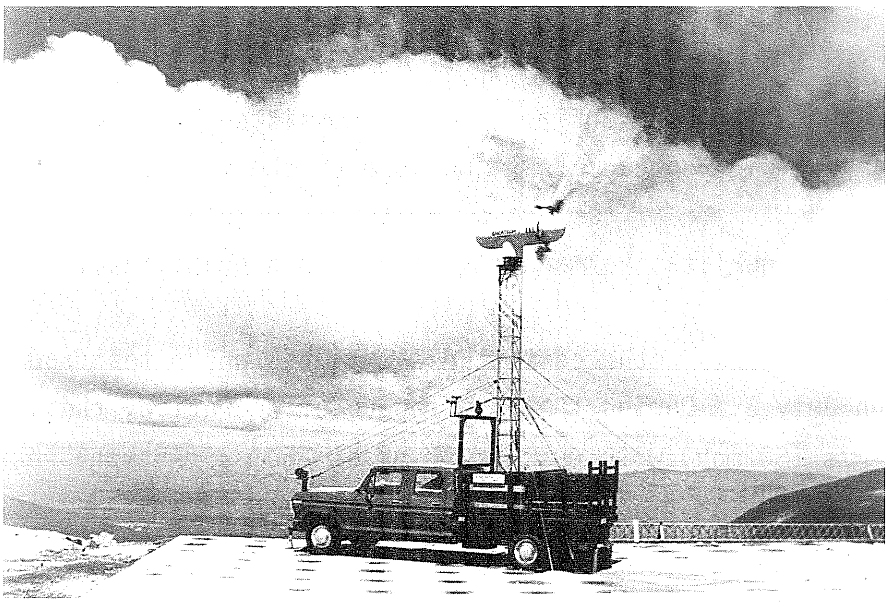


FIGURE 26: TESTING AT MOUNT WASHINGTON

During the first few hours, when normal operation occurred, the following observations were made: 1) Start-up was noticeably sluggish. Wind speeds in excess of 7 m/s were required for start up. 2) Slight one-per-revolution vibrations were noticed. 3) Tracking appeared smooth and accurate even in light winds. 4) Rotor speed limited normally. 5) Predicted power was produced.

Inspection of the machine showed that the brake cable was binding. This caused the rotor brake to be partially applied during operation, accounting in part for the high wind necessary for start-up. It was later discovered that 12 ounces of balance weight was missing from the rotor. This weight had been removed during the demonstration installation of the machine on a tower in June. It was discovered during the installation that the balance weight interfered with the attachment of the fiberglass spinner to the hub. The weight was removed in order to expedite the demonstration and, due to an oversight, was not replaced before the tests on Mount Washington were initiated. The missing balance weight explained some of the vibration that was observed during these tests.

After several hours of operation, a major failure occurred that eventually halted the testing. A gearbox failure caused the decoupling of the rotor from the generator and the brake. Without the load from the alternator, the rotor speed increased and limited at a higher value. The brake could not be used to stop the rotor. The level of vibrations was greater than before. When yawing, the machine vibrations were severe. After several hours of operation with vibration, the nacelle fasteners loosened, allowing the nacelle to come off. The nacelle flew through the spinning rotor and was irreparably damaged. Eventually the leveling bolts loosened, causing the machine to "tip" into the tower, destroying the blades and halting the machine.

Following the testing, the machine was disassembled and the components were inspected for damage. The results of the inspection were: 1) When the gearbox was opened and inspected, all the fastening cap screws inside the gearbox were found to have come loose and were lying at the bottom of the housing. The gears were stripped and the input and output shafts "decoupled" from each other. The failure of the gearbox explained the failure of the machine. The failure of the loads to slow the machine and the failure of the rotor brake to stop the machine are directly attributable to the decoupling of the gearbox input and output shafts. In addition, the loosening of the screws caused the input shaft, to which the rotor is attached, to be unsupported, explaining the excessive rotor vibrations encountered. The gearbox used for these tests was repaired. In addition, the gearboxes used after this test were disassembled and all fasteners were secured with Loc-Tite. 2) Fasteners on the nacelle bracket, fasteners on the alternator, and the leveling bolts had vibrated loose. As extended operation had not been planned, lock washers had not been used on the leveling bolts. However, since even fasteners with lock washers had vibrated loose, the entire fastening system required re-evaluation. Self-locking nuts and flat washers have been specified for bolts. Screws will be secured with locking washers and Loc-Tite or safety wire. Following the Mount Washington tests, the unit was repaired and shipped to Rocky Flats for buyer testing.

6.5 Buyer Testing

In September of 1979, Prototype 1 was shipped to Rocky Flats for testing. Following component testing the unit was installed on a Rohn 45GSR tower in December of 1979.

6.6 Start-Up Tests

In October of 1980, Enertech was awarded a fixed-price contract to improve the start-up characteristics of the wind machine. The requirements of the contract were as follows:

1. The machine shall start up at a sustained wind speed of 4.5 m/s as verified by the contractor during truck tests at the contractor's location.

2. Power output shall be a minimum of 2kW at 9 m/s (sea level equivalent).
3. The control system shall limit the rotor speed in high winds as verified by contractor truck testing to wind speeds up to 50 mph (at contractor's location).

During controlled-velocity tests of the wind machine on a test truck facility on April 22 and 23 of 1981, the required performance was recorded.

By decreasing the drive train break-away torque (the torque on the low-speed gearbox shaft needed to start the drive train turning from a stop) and by increasing the available rotor torque, the start-up requirements were met. While changes were made to both the drive train and rotor, the decrease in the drive train friction was the major factor in decreasing the start-up wind speed. The drive train originally had a break-away torque of 40 inch-pounds and the wind machine started in a 6 m/s wind. The modified drive train had a break-away torque of 19 inch-pounds and the machine started in a 4.5 m/s wind. Of the 19 inch-pounds of break-away torque, 5 inch-pounds were needed to turn the gearbox alone.

The gearbox, as received from the manufacturer, had a shielded bearing on the high-speed shaft. Because the internal components of the gearbox operate in an oil reservoir, it was decided that it was not necessary to use a shielded bearing, so an open bearing was substituted. This change should not alter the reliability of the gearbox or affect its performance except by reducing the friction.

The alternator was modified by replacing the open bearings with sealed bearings, removing the teflon shaft seals, and removing the grease from around the shaft. Because there have been no problems with the sealed bearings in the motors for the Enertech 1800 wind machine, no problems are expected for the sealed bearings used on the 2kW machine. Until the alternators with the sealed bearings are field tested, however, the effect of this modification on reliability will not be known.

The gearbox and alternator were motor driven for 500 hours on a test stand to reduce the drive train friction. Little additional reduction of the break-away torque resulted after the first 150 hours.

To increase the rotor torque available, the blades were redesigned and the initial blade root pitch was increased from 9 to 9-1/2 degrees. After comparing the performance of the new blades with the performance of the old blades, the following conclusions were reached: 1) With each set of blades, start-up occurred at 4.5 m/s. 2) At the rated wind speed the redesigned blades produced 3.0kW while the original blades produced 2.2 kW. 3) The maximum power output was 3.7 kW with either set of blades. The power curves for both sets of blades are shown in Figure 27. Because the redesigned blades produced more power in light winds, this design will be recommended for future units.

During truck tests, the wind speed, alternator speed, voltage, and current were monitored on a chart recorder. Only three of the four variables could be recorded at any time. After the tests, the data was analyzed and about 150 data points were chosen from the chart. To determine the power curve, the wind speed range of the test was divided into increments and the power output at the mid-point of each increment was taken as the average power output of the data points falling in that increment.

Rotor motion began when the winds were 3.6 m/s. By maintaining the truck speed between 3.6 and 4.5 m/s (8-10 mph) for 2.8 minutes, the rotor speed increased until power was produced. This test was repeated three times on a calm day. Figure 28 shows a tracing of the chart recording for a start-up test.

Power output at 9 m/s (20 mph) averaged 3.0 kW. Figure 29 shows a tracing of the chart recording for the tests conducted at the rated wind speed.

To test the speed limiting characteristics of the rotor, the truck was driven at speeds up to 22 m/s (50 mph) and the rotor speed was monitored. Figure 30 shows the performance of the rotor during a high-wind test. The maximum power recorded was 3.7 kW in a 13.5 m/s wind.

In conclusion, the requirements of the fixed-price contract were met and as a result of this program the performance of the machine was improved.

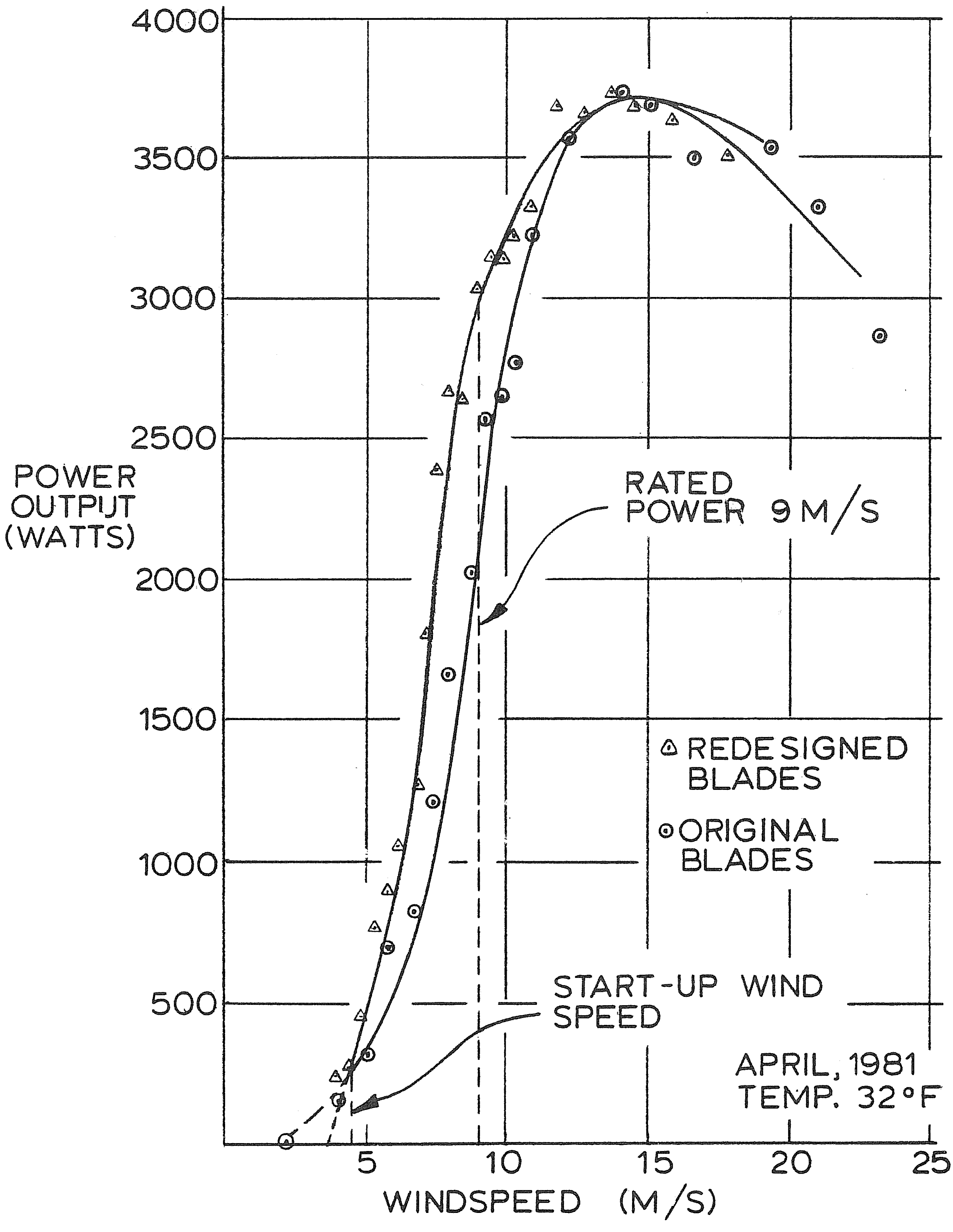


FIGURE 27: POWER VS. WINDSPEED

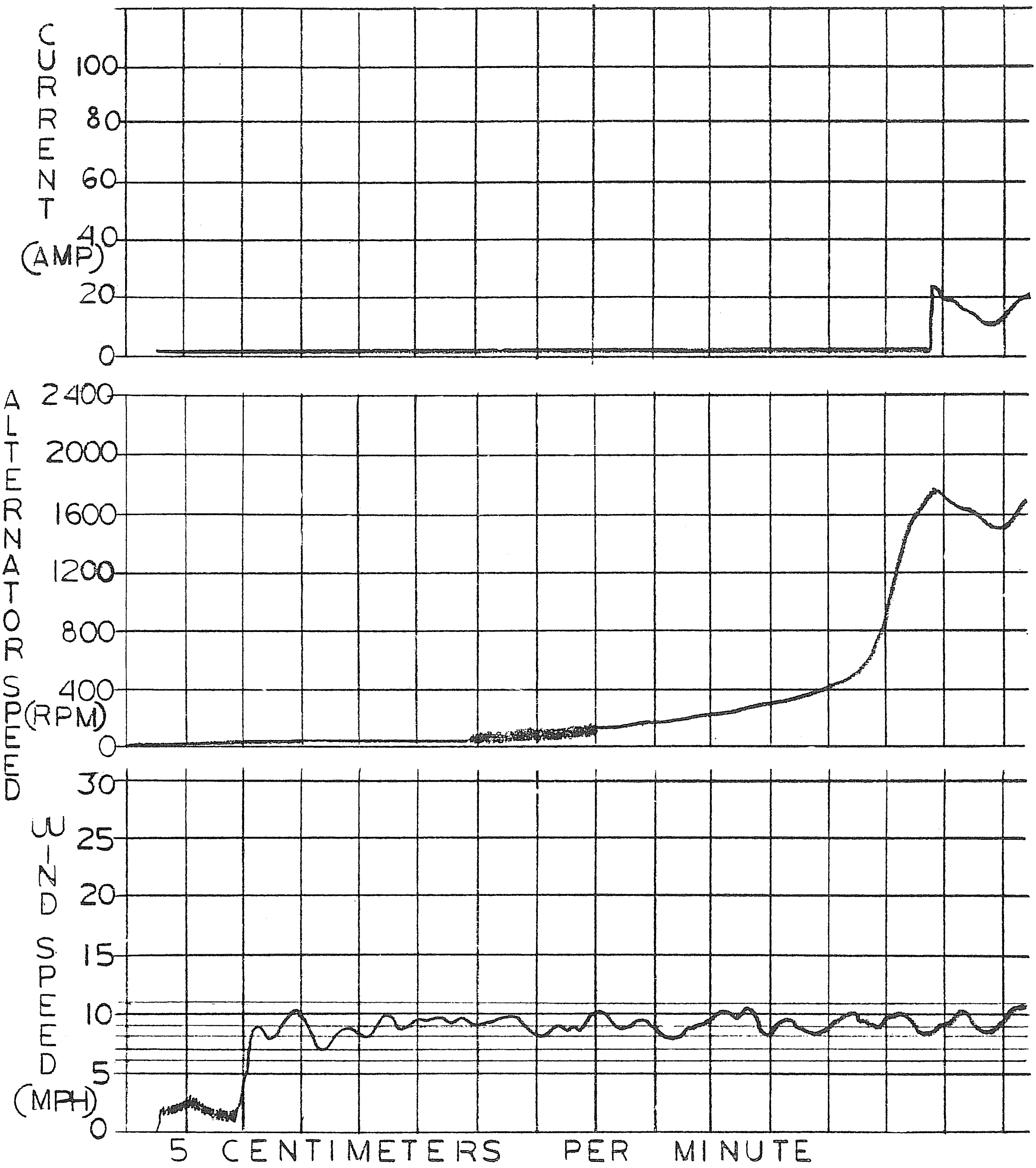


FIGURE 28: ROTOR START-UP RECORDED DURING TESTING

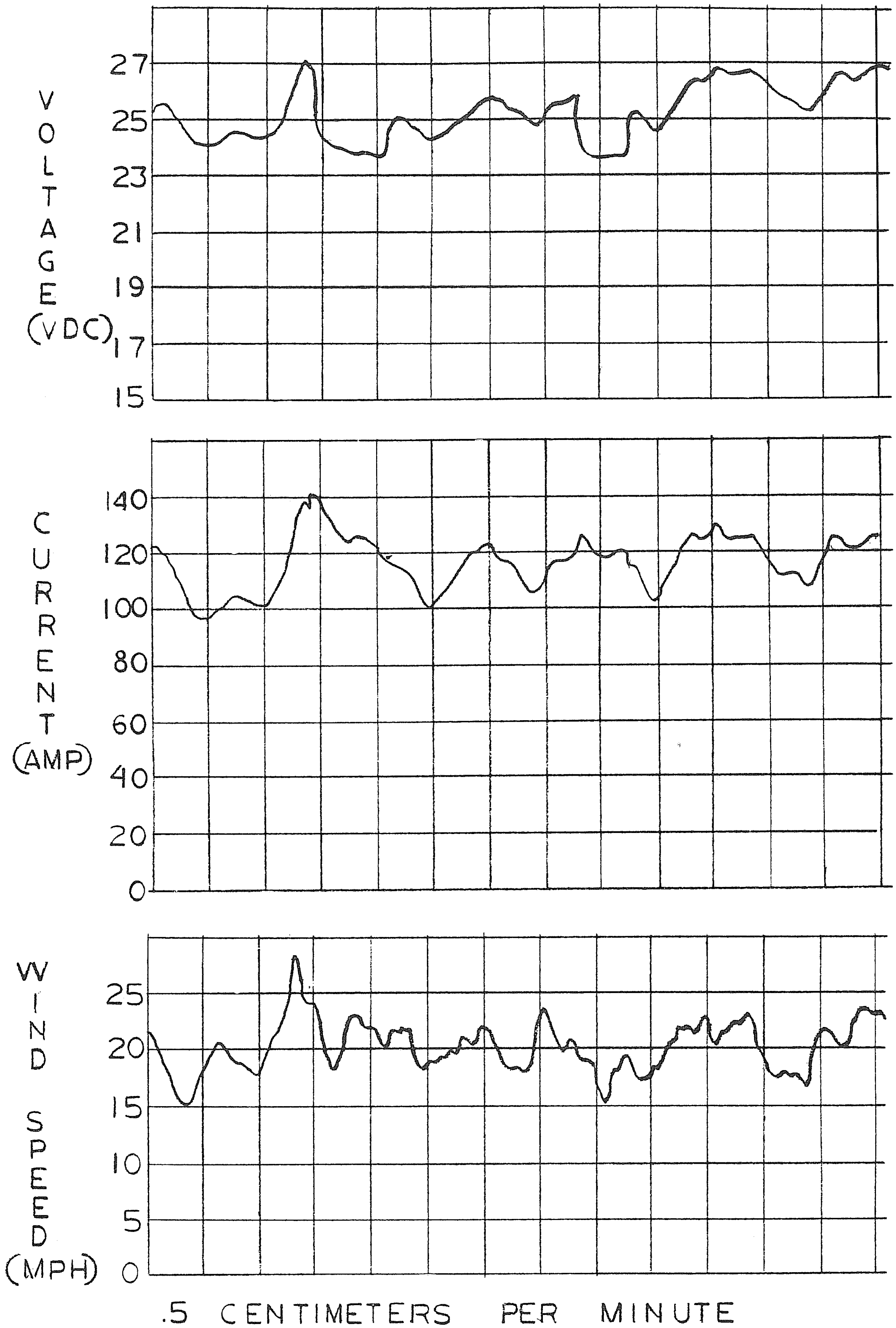
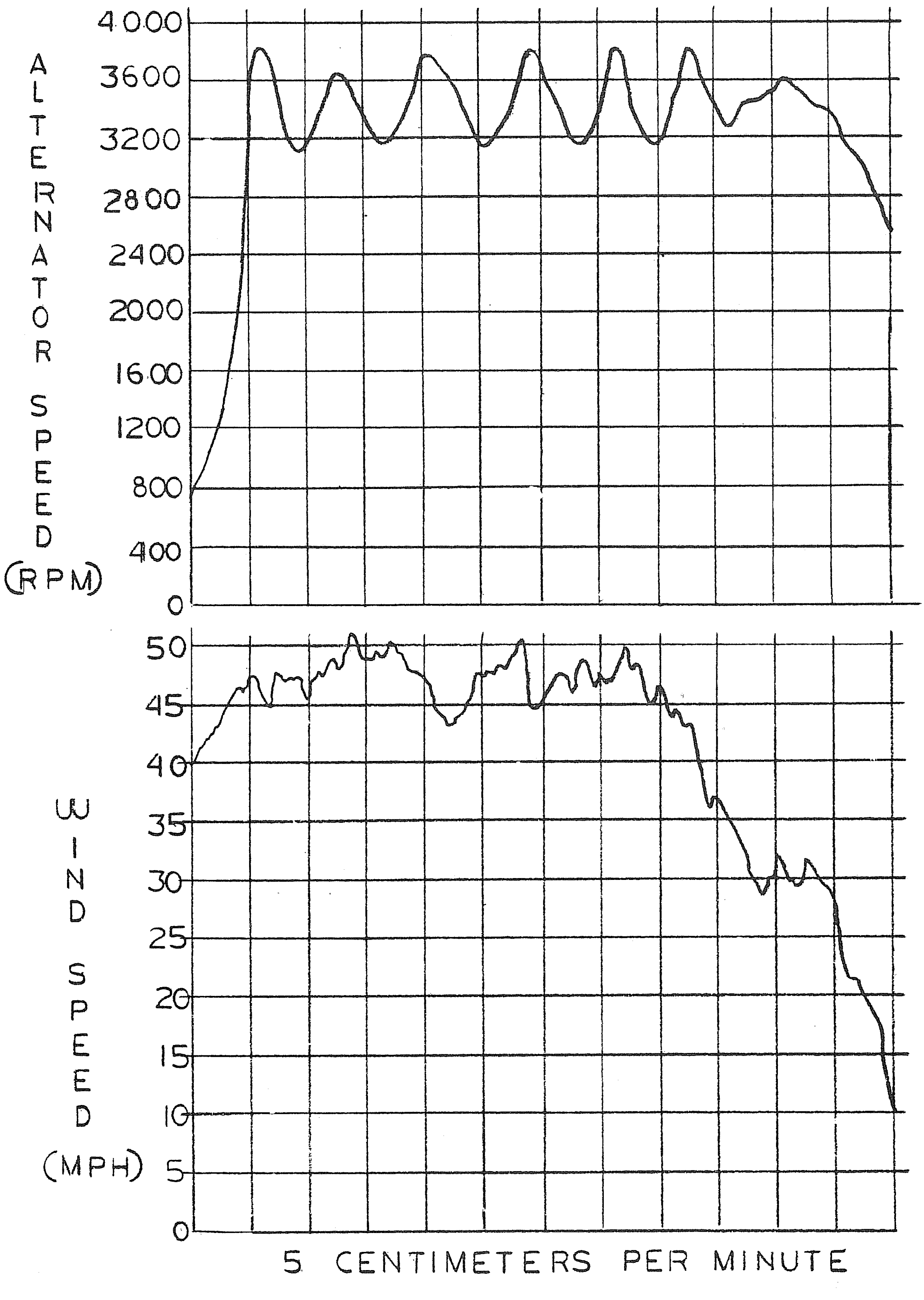


FIGURE 29 : POWER OUTPUT AT 9 M/S (20 MPH) RATED WIND SPEED



GEAR RATIO: 10.89:1

FIGURE 30: ROTOR SPEED LIMITING CHARACTERISTICS

7. Economics

The unit price for a 2 KW High-Reliability Wind Machine based on 1000 production units is $2982, as shown in Figure 31. The dollar figures in this analysis are given in 1978 dollars. To determine energy costs, in dollars per kilowatt-hour, the total estimated machine cost is compared with the energy produced by the machine. For simplicity, this has been calculated on an annual basis.

Figure 32 summarizes some cost information for the system and shows a sample calculation of the cost of energy produced by the system. The estimated FOB cost of the machine in production is $2982 and the estimated FOB cost of the tower is $890 for a total of $3872. Multiplying this times 1.5 yields an estimated retail price of $5808. The cost of installing the tower and wind machine at accessible sites and in volume is estimated at $800, for a total installed price of $6608 (in 1978 dollars).

From the total installed price, the cost of the energy produced by the system can be calculated as shown in the sample calculation in Figure 32. Here an annual fixed charge rate of 0.087/year has been used.* For annual operation and maintenance, the cost of one person-day per year plus certain supplies has been used, amounting to about 2.5% of the installed cost per year.

Using the NASA/Lewis wind speed distribution, the annual kilowatt-hours which the system will produce have been calculated for three average annual wind speeds. The average annual wind speeds considered were 12 mph (5.4 m/x), 15 mph (6.7 m/s), and 18 mph (8 m/s) at a height of 30 feet (9.1 meters). The wind speeds seen by the wind machine were corrected for tower height. At 12 mph, AKWH equals 18,200.

Finally, using the above values, the cost of energy can be calculated. In the example shown, the cost of energy has been calculated for a "base case" at 15 mph average wind speed; in this example, the cost of energy is $.054/kwh.

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* The fixed charge rate cost of energy calculation method used in this report was specified by Rockwell International to allow comparison between this machine and others developed under DOE funding. The reader should be aware that life cycle costing gives a more accurate cost of energy calculation. A good introduction to this method--as it applies to wind systems--can be found in SWECS Cost of Energy Based on Life Cycle Costing, W.R. Briggs, Rocky Flats Wind Systems Program, RFP-3261, May 1980 (available form NTIS).

Figure 33 shows a sensitivity diagram which illustrates the effect of changes in each of the base case parameters on the overall cost of energy. Windspeed clearly exerts the strongest influence on cost of energy, while factors such as annual maintenance and operation cost exert less effect - as long as the site is reasonably accessible.

FIGURE 31

UNIT PRICE ESTIMATES

Production Rate - 1000 units per year

Cost Element	Description			TOTAL
1. Direct Materials				
a. Purchased Parts	Gearbox	325		
	Hub Bearings (4)	12		
	Yaw Bearings (2)	6		
	Sliprings and Brushes	90		
	Rotor Brake	40		
	Ammeter	25		
	Voltmeter	20		
	Lightning Protection	25		
	Control Box	15		
	Lead-in wire, 100'	83		
	2" Metal Conduit, 40'	62		
	TOTAL	703		
a. Subcontracted Items	Alternator	370		
	Flange Adaptor	30		
	Nacelle	75		
	Blades	97		
	Hub Parts	256		
	Frame Parts	85		
	TOTAL	913		
c. Raw Materials	Fasteners	15		
	Electrical Wire	2		
	Electrical Connectors	6		
	Labels	2		
	Paint	2		
	Wood (crating)	15		
	TOTAL	42		
Total Direct Materials			1658	
2. Material Overhead (12%)			199	
	SUBTOTAL			1857
3. Direct Manufacturing Labor		108		
4. Manufacturing Labor Overhead	Overhead @ 100%	108		
5. Direct Engineering Labor		18		
6. Engineering Labor Overhead	Overhead @ 90%	16		
7. Interest Expense		170		
	SUBTOTAL			2277
8. General, Administrative, and Selling Expense	@ 20%	455		
	SUBTOTAL			2732
9. Profit		250		
	Factory Price	2982		
	Price per kW rated	1491		

FIGURE 32

COST OF ENERGY CALCULATION

Basic Method of Calculation:

$$COE = \frac{(IC)\ (FCR) + AOM}{AKWH}$$

Where: COE = Cost of Energy

IC = Initial installed system cost (dollars)

FCR = Annual Fixed Charge Rate = 0.087/yr. (see note, page 63)

AOM = Uniform Annual Operation and Maintenance Cost (dollars)

AKWH = Total Annual Kilowatt-Hours Produced

Calculation of IC:

Cost of 1000th Wind Machine, FOB cost x 1.5

2982 x 1.5 = 4473

Cost of Tower, FOB cost x 1.5

890 x 1.5 = 1335

Cost of System Installation, including excavations, concrete, labor (no batteries, etc.) = 800

IC = Total of Above = 6608

Calculation of AOM:

Allowable Maintenance (1 person-day per year) = 150

Parts-lubricants, paint, etc. = 20

Total AOM = 170 = 2.5% (IC)

Calculation of AKWH:

AKWH has been calculated for 3 windspeeds using the NASA/Lewis distribution:

AVE. Windspeed	AKWH
12 mph	8,400
15 mph	13,600
18 mph	18,200

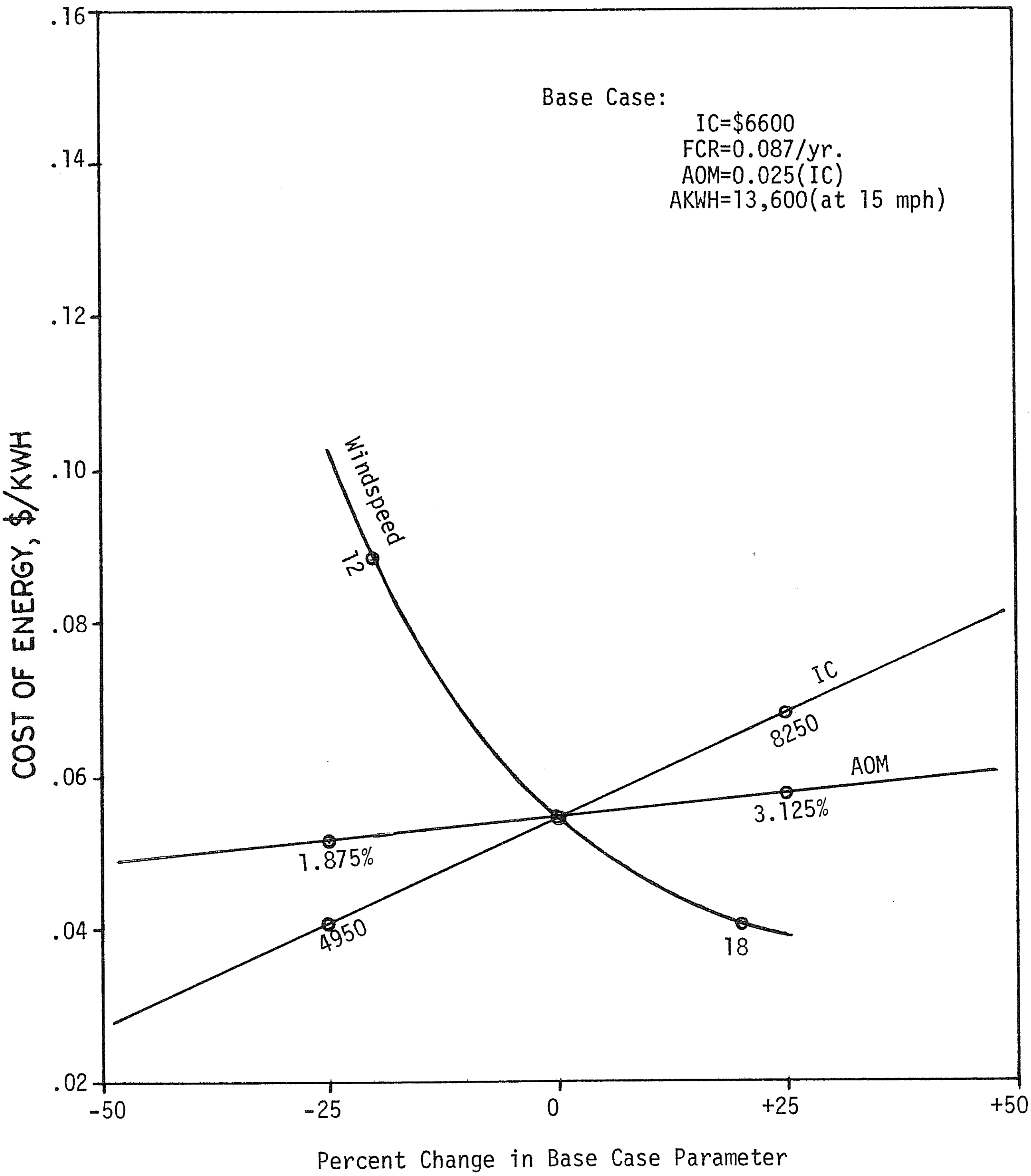
Sample Calculation: Cost of Energy

$$COE = \frac{6600\ (0.087) + 0.025(6600)}{13,600}$$

$$= \$.054/kWh$$

FIGURE 33

SENSITIVITY OF COST OF ENERGY - ENERTECH 2KW WIND MACHINE



8. Summary

The work during the Phase II portion of this contract centered on the fabrication and testing of three prototype wind machines. The fabrication of the wind machines consisted primarily of the assembly of purchased components.

Prototype 1 was tested at the Enertech test facility, at Mount Washington, on a test truck, and at the Rocky Flats test facility. The testing effort was extremely valuable. After each test, design changes were specified and the unit was improved. During testing at Rocky Flats, there were two remaining major problems with the design: poor start-up characteristics and wind machine/tower interactions. Following contractor improvements to Prototype 3, the wind machine started in 4.5 m/s winds, an acceptable wind speed.

Figure 4 in Section 1 outlinea the Phase II tasks for the program. Except for several tasks which were deleted from the program by agreement between the buyer and the contractor, all work requirements for Phase II will be completed with the completion of this Phase II report.